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SIXTY-SEVENTH YEAR

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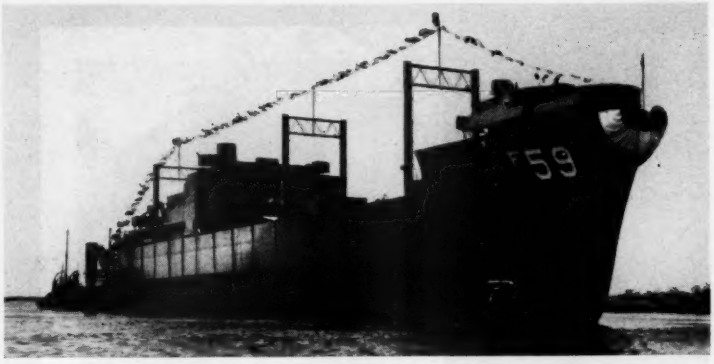

U.S.S. Rigel after launching from the ways of The Ingalls Shipbuilding Corporation, March 15, 1955

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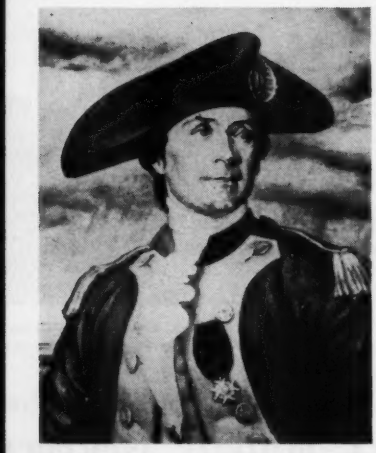

U. S. S.
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John Paul Jones, one of America's earliest and most colorful heroes, bequeathed a legacy rich in fighting spirit to our Navy. It is fitting that the latest addition to our destroyer fleet should bear his proud name.

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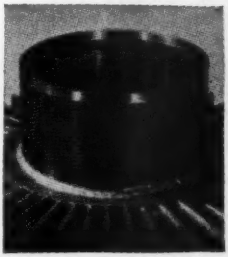
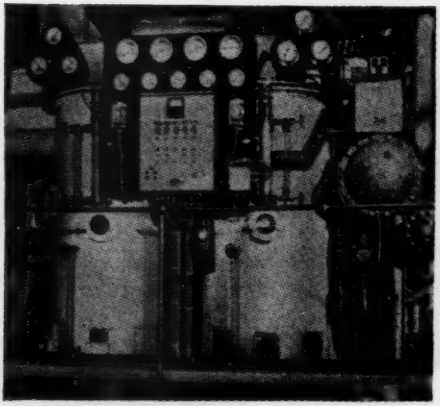


Monel heater section—This welded Monel heating section must endure heat from low pressure steam, corrosion from boiling sea water, as well as thermal shock. The inside is a smooth, satin-finish.



Long life—The Maxim Silencer Company uses corrosion resisting Monel in building this sea water still for long life and low maintenance. One of two on the new *S. S. Andros Venture*, it can distill 12,500 gallons a day.

Why this Maxim Evaporator needed

No Descaling for 750 hours!

See this big distiller. It is one of the first of its kind to be used aboard a commercial vessel. It's one of two 12,500 gallon-a-day Maxim Evaporators on the new 28,000 ton *S.S. Andros Venture.*

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← **Shop training scene** which shows an apprentice shaping a small die on a shaper, indicates how students are thoroughly trained.

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Shown here is a Crane 600-pound Pressure-Seal gate valve, toggle-operated with air motor. It's typical of the many valves Crane is supplying for catapult service on Navy carriers both new and conversions.

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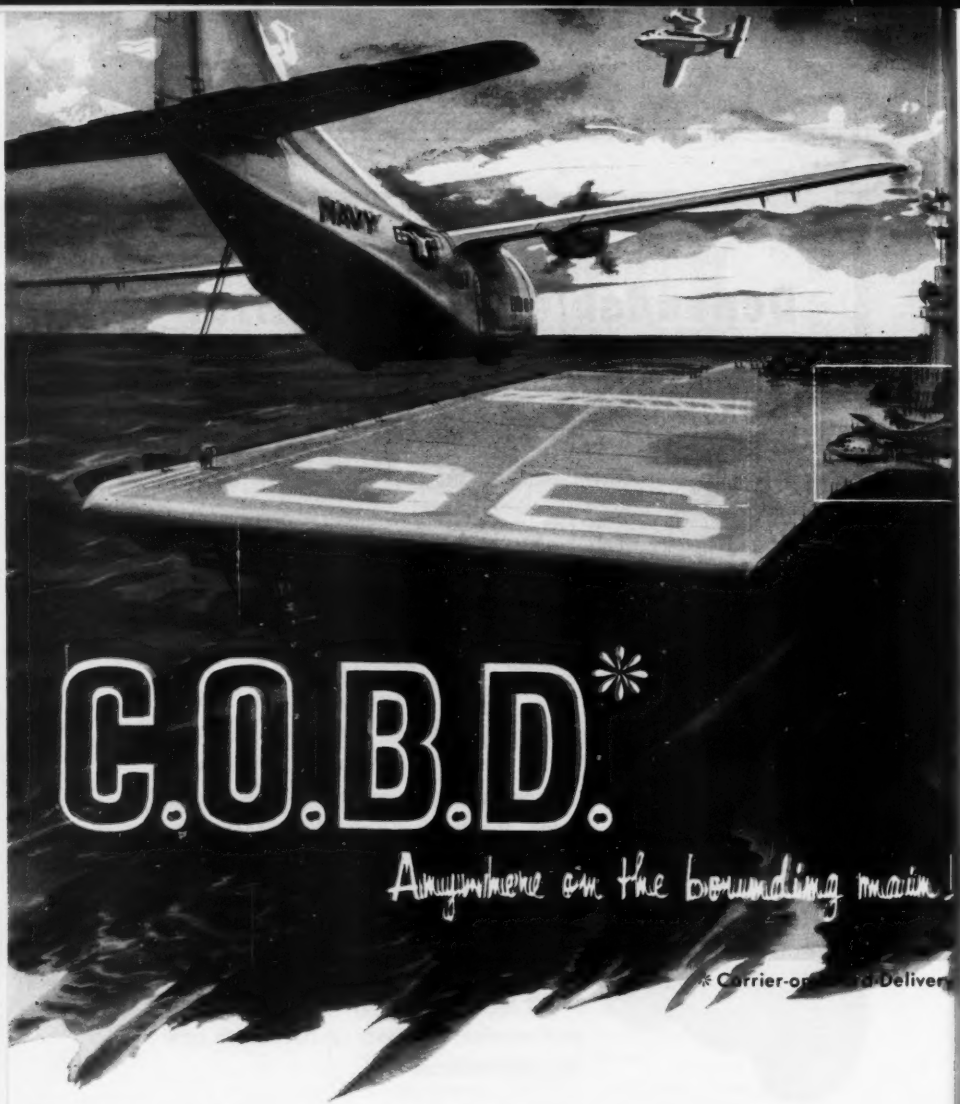

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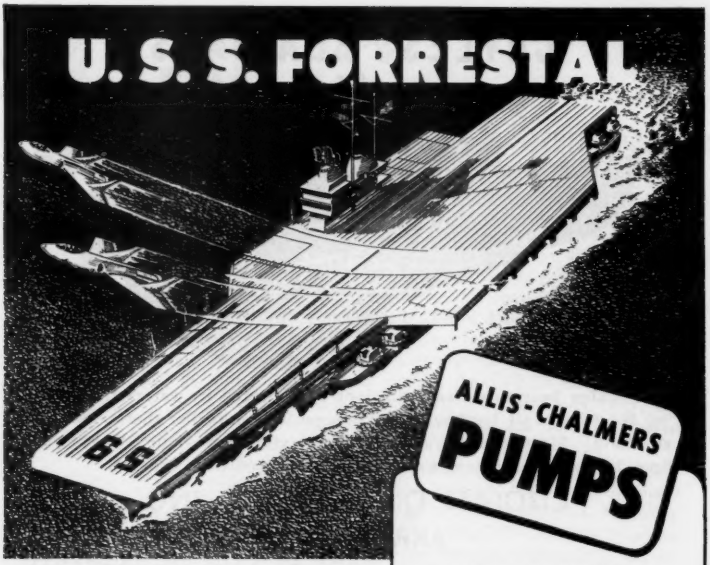

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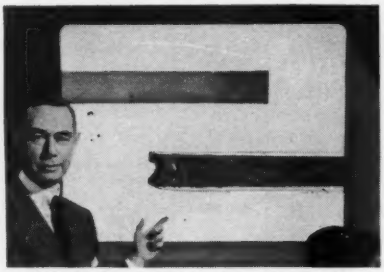

High-velocity, turbulent sea water damaged these Admiralty tubes. Top, flow over pebble or shell left small scar. Bottom, tube inlet is badly corroded and eroded.

VELOCITY: how serious tube damage from this cause can be avoided

High velocity is a big factor in causing turbulent flow of the cooling water. This in turn may lead to rapid inlet-end corrosion of condenser tubes. You can smooth out the flow of water by using deep water boxes and by flaring the inlet ends of tubes. And you should do away with water obstructions.

Where a tube is partially blocked by a pebble, shell or other debris, local high velocities can occur. It is highly desirable to inspect and clean tubes as frequently as possible.

Velocity is only one factor affecting tube life. Publication B-2, "Anaconda Tubes and Plates for Condensers and Heat Exchangers," discusses *all* the important operational factors, and gives much other useful information. *The American Brass Co., Waterbury 20, Conn.* 54106A

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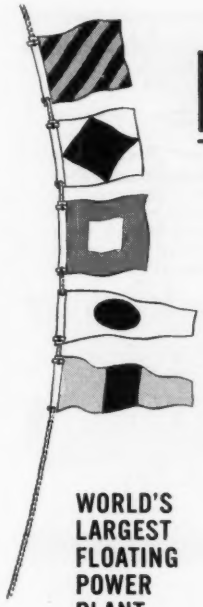

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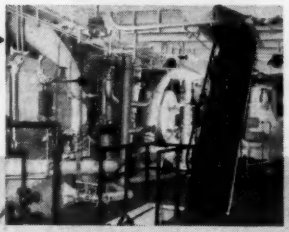
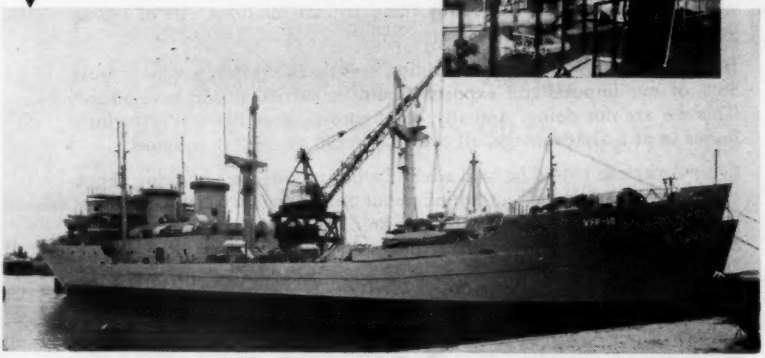
Designed to equal or exceed the efficiency of a shore-built plant of equal output, the YFP-10 can go from a "cold ship" to full power output in four hours. Operation at normal load can be continued for 30 days without refueling.

To promote overall economy, three top-fired Type-D Foster Wheeler steam generators of advance design have been installed. These units are capable of delivering 123,000 lbs per hr at 600 psig, 825F, and are complete with Foster Wheeler superheaters, economizers and air heaters. *Foster Wheeler Corporation, 165 Broadway, New York 6, N. Y.*

View showing top-fired Type-D Foster Wheeler steam generators. ▶

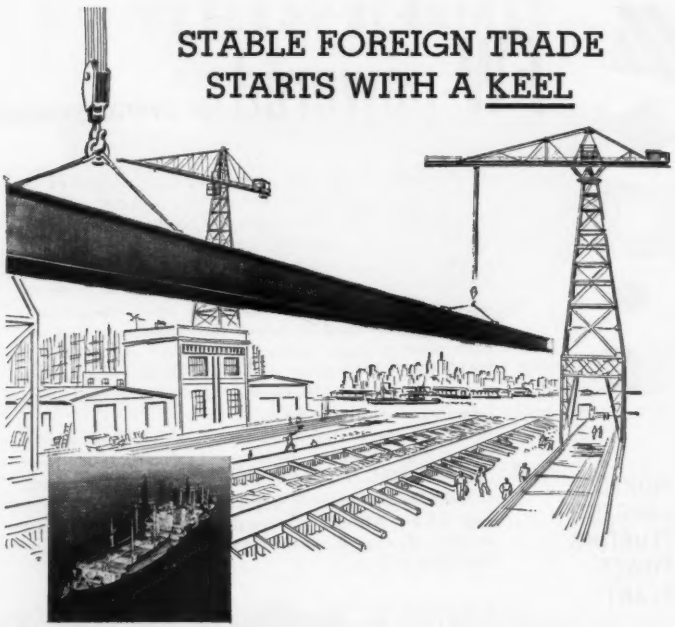


U. S. Navy YFP-10, turbo-electric generating plant, capable of delivering 34,000 kw at 13,000 v. Built for the Bureau of Yards and Docks by the Gibbs Corporation, Jacksonville, Fla. Power plant designed by Reynolds, Smith and Hills, Engineers, Jacksonville, Fla. ▼



FOSTER WHEELER

STABLE FOREIGN TRADE STARTS WITH A KEEL



It's a point of view pursued by almost every maritime nation in the world. They build ships as we build roads—steadily, year in, year out. Shipbuilding programs in many nations the world over are *part of their national economy.* World trade, aside from keeping those engaged in it alive, also is essential to healthy international relations. All of which you probably already know.

But did you know that expert opinion says that for safety's sake at least 50% of our imports and exports should be carried in our own ships? This we are not doing. Actually, the figure is more like 29% — which leaves us at a disadvantage, all over the world as well as at home.

Every American would be well advised when talking and thinking about our foreign trade to talk and think about our Merchant Marine and how strong it should be and how strong it isn't. It's a fact that concerns us all as long as we do business outside our borders.

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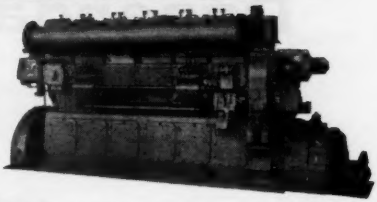

truly advanced engineering, has enabled Cooper-Bessemer Diesels to do a notable job in Navy and Coast Guard vessels from the biggest combat ships to relatively small patrol boats. Yes, you can always count on Cooper-Bessemers for Diesel performance at its finest.

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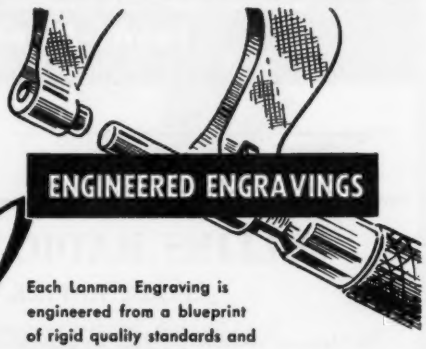

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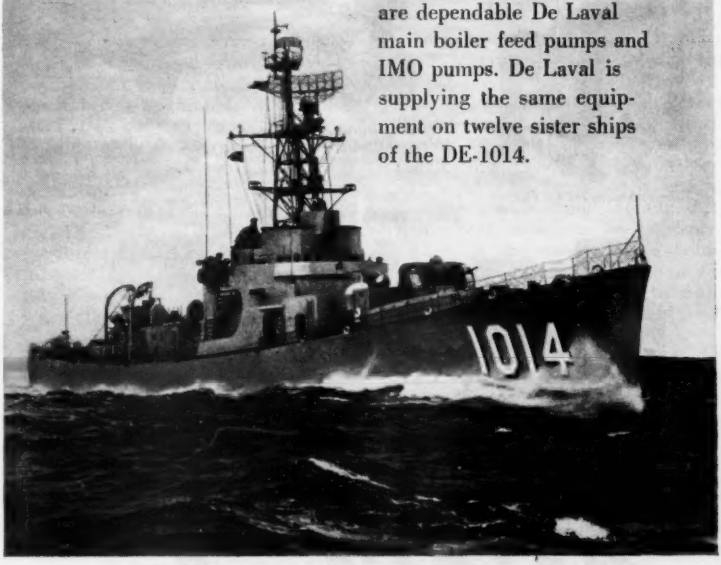

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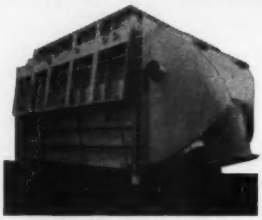
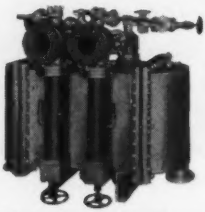


Dual Bank Steam Condenser

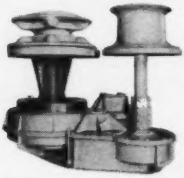
Automatic Tensioning Hoist — Airplane Crane

Electric Hydraulic Deck Winch



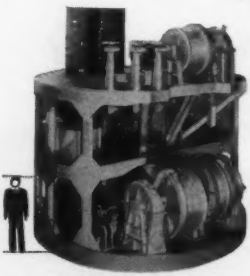
Tubejet Steam Jet Air Ejector

Electric Warping Capstan

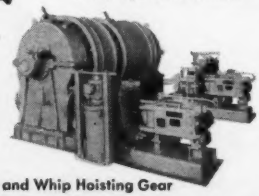


Electric Hydraulic Windlass

Warping Winch



Boat and Airplane Crane



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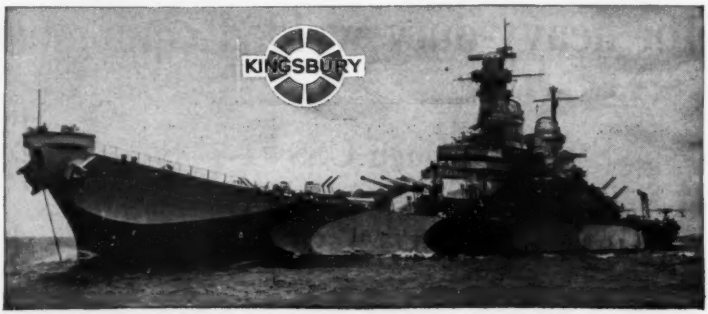

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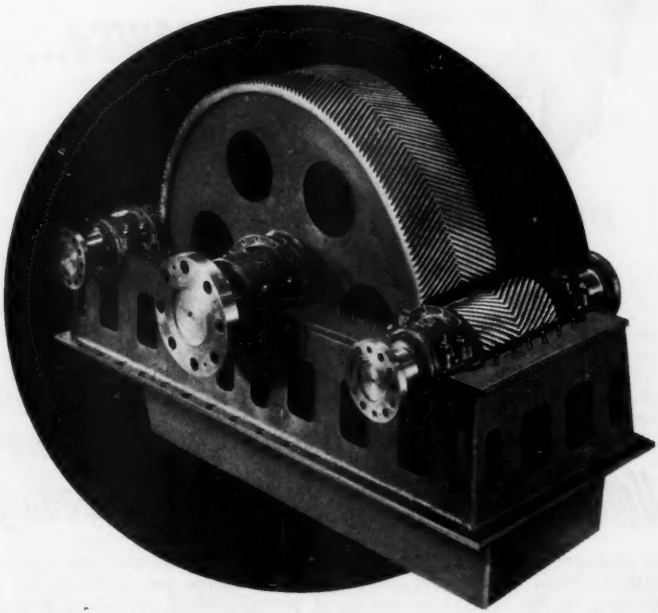

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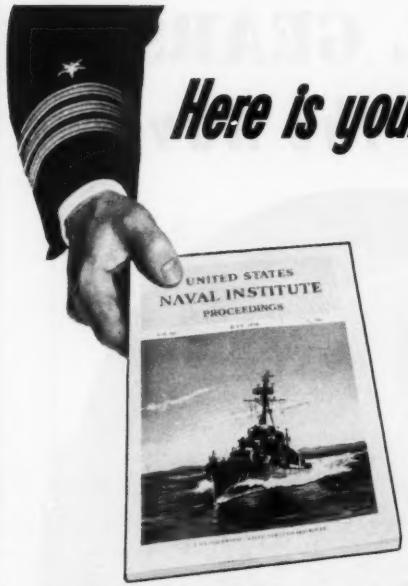

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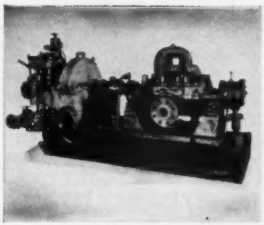
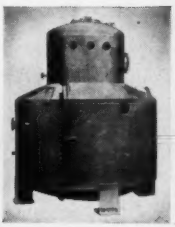
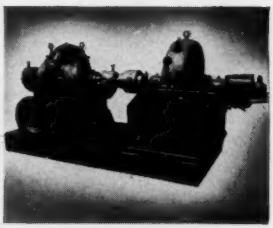

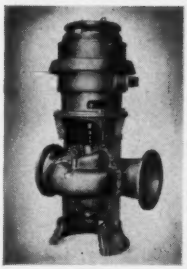
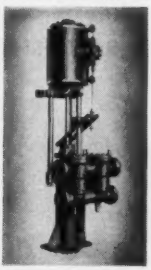
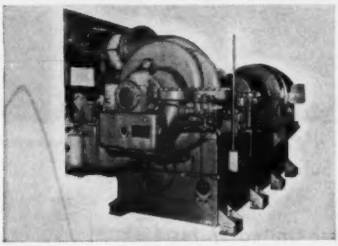
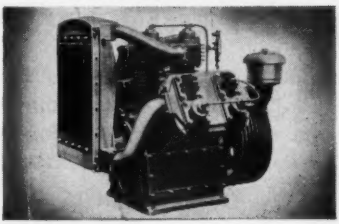

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JOURNAL

OF

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Vol. 67 AUGUST, 1955 No. 3

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SECRETARY'S NOTES

THE BRAND AWARD

Last year Mrs. Charles L. Brand instituted the Brand Award in commemoration of her late husband, Rear Admiral Charles L. Brand. This year, on June 10, 1955, this award, a year's subscription to the JOURNAL, was presented to

Lieutenant C. F. Baker, U. S. Coast Guard

Lieutenant Baker stood highest in this year's graduating class in Course XIII-A, Naval Construction and Engineering, at the Massachusetts Institute of Technology.

SIMPLIFICATION OF DRAWING PRACTICES

The Bureau of Ships, Department of the Navy, has initiated a project to simplify engineering drawing practices used to delineate ships and ship components. The Bureau has available to it the methods and techniques used by the General Electric Company and the Radio Corporation of America; however, it is desired to accumulate as much information as possible from others who have worked in this field.

Any member knowing of papers, systems, or techniques covering Simplified or Functional Drafting is requested to communicate with the San Francisco Naval Shipyard, to whom this project has been

assigned for detailed development. Communications should be addressed to:

Commander, San Francisco Naval Shipyard
Attention: Design Division, Code 242P
San Francisco 24, California

Engineering Abstracts

Commencing with this issue a new feature will be included in each Journal,—a compilation of abstracts of those engineering articles and papers of particular interest to members of the Society. The idea, originated by John W. Sawyer, is considered to be a worthwhile addition to the custom of many years' standing of reprinting from other publications articles dealing with naval engineering. Because of printer's deadlines, the reprints have not been as current as your editorial staff would like them to have been, and space has always limited the number of reprints actually used. The "Engineering Abstracts," it is hoped, will partially alleviate both of these faults.

Mr. Sawyer, Head Civilian of the Gas Turbine Branch, Bureau of Ships, and member of the Society's Council, is not only the originator of the idea—he will also be the compiler. For a biographical sketch readers are referred to page 565 of this issue.

The comments and suggestions of Society members regarding this new feature are solocited.

PLASTIC COVERS FOR PROPULSION GEARS ON U. S. NAVAL SHIPS

R. C. JOHNSON and J. W. SAWYER

THE AUTHORS

R. C. Johnson was a Marine Engineer in the Special Projects Section of the Turbine and Gear Branch of the Bureau of Ships. This section is concerned with Research and Development Work for the improvement of marine reduction gears and associated equipment. Prior to employment at the Navy Dept., he was assistant to the Chief Engineer at Briggs Filtration Co.; a heating and air conditioning sales engineer for McCrea Equip. Co., and a field engineer for steam turbine installation with General Electric Corp. Earlier he was a design engineer for the Reynolds Metals Co. and the Solvay Process Corp. He completed a specialized course in Mechanical Engineering at the University of Pittsburgh in 1940, entered the Air Force and served as a fighter pilot from 1943 to 1945. At present he is in the Value Engineering Branch of the Bureau of Ships.

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INTRODUCTION

Reductions in weight of most machine components, while at the same time retaining the utmost in reliability, are usually of prime interest to designers, manufacturers and the users. This subject of weight saving is of particular interest where pounds saved in one place can result in improved performance. This is especially true in many naval applications. The Bureau of Ships has recently installed a plastic cover, of sandwich type construction, with polyester shell and cellular cellulose acetate core, Fig. 1, on a propulsion reduction gear case, Figs. 2, 3 and 4, of the USS *Fitch*, DMS25. This cover is a non-strength member of the gear case but is expected to withstand the usual treatment its steel counterpart would receive in an engine room.

This installation is one step beyond the laboratory test, (1) on a plastic cover which showed outstanding promise. The weight of conventional steel covers for all propulsion gear cases on a DMS is about 3,240 pounds. Plastic covers of the type recently installed weigh approximately 993

pounds. This results in a weight saving per ship of some 2,244 pounds or 70 per cent over the steel design.

Light weight covers offer additional attraction in that they are more easily handled by maintenance personnel. Compare the ease of lifting a 249 pound section of a plastic cover against that of hauling up a section of a steel cover weighing 810 pounds. Additional details on weights of various covers are shown in Table I.

The description, specifications, materials, manufacture, test and installation with suggestions on construction of future on plastic gear case covers are discussed in the following paragraphs.

Table I—Dimension and Weight

	Dimensions (In.)				Weight (Lb.)
	Length	Width	Height	Thickness	
Conventional Steel Construction	83	68	19-½	3/16	810
Plastic Nr. 1 (Single Skin, Polyester-Glass)	83	68	19-½	0.20-0.35	237
Plastic Nr. 2 (Sandwich Const. Polyester CCA core)	83	68	19-½	15/16"	249
Plastic Nr. 3 (Modified Phenolic Resin)	83	68	19-½	0.20-0.35	250

DESCRIPTION

The reduction gear selected for the operational evaluation of a plastic cover was the port, low pressure unit of the USS *Fitch*, DMS25. The installed plastic cover for this unit is made of two sections, one forward and one aft, which is similar to the original steel cover. These sections are bolted together, Figs. 3 and 4, to form one cover, which has the following dimensions: length 83 in., width 68 in., height 19½

Figure 1. Gear case cover of sandwich type construction, with polyester glass laminate shell and cellular cellulose acetate core.

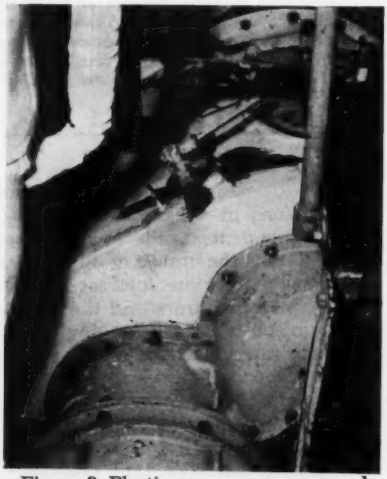

Figure 2. Plastic gear case cover, sandwich type construction, installed on USS *Fitch*, DMS25. End view looking aft.

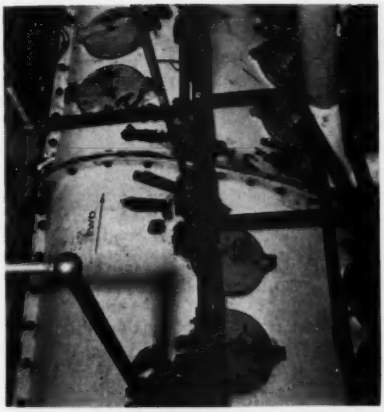

Figure 3. Plastice gear case cover, sandwich type construction, installed on USS *Fitch*, DMS25. View from position above gear case.

in., thickness 15/16 in. The cover contains ten flanged inspection ports 9 in. diameter, and ten thermometer connection openings 3 in. diameter. Rosan type threaded metal inserts were used to permit attaching inspection opening covers, and thermometers. Eyebolts were installed to assist in handling the cover during placement and lifting. All flanges and inspection port bosses are of solid polyester-glass laminate. Two other covers were manufactured and given laboratory tests along with this unit. The specifications for the three covers that were designed, manufactured and laboratory tested are described in the following paragraphs.

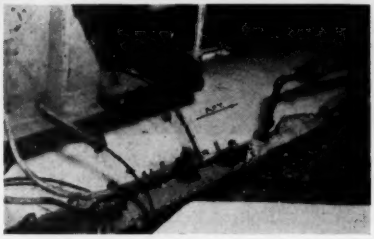

Figure 4. Plastic gear case cover, sandwich type construction, installed on USS *Fitch*, DMS25. View from inboard side.

SPECIFICATIONS AND QUALIFICATION TESTING

Three different designs were selected for manufacture and tests. They are of the following construction:

1—Single skin fiber-glass reinforced polyester resin, Fig. 5.

2—Sandwich type, polyester glass laminate shell with cellular cellulose acetate core, Fig. 1.

3—Single skin modified phenolic resin, reinforced fiber-glass, Fig. 7.

The object of producing three different covers was to determine the design, methods and materials that would give the best overall performance.

Materials were required to withstand impingement of lubricating oil without loss of strength or rigidity. They were to be fire resistant to the extent of being self extinguishing. Resins were to be reinforced with fibrous glass mat or cloth. The resin content was not to exceed 70 percent by weight. Water and oil absorption was to be insignificant. Resin starved

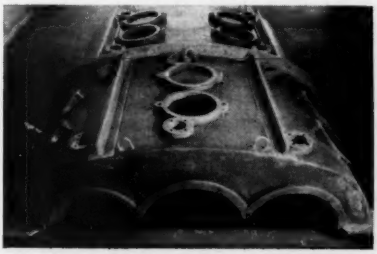

Figure 5. Single skin fiber-glass reinforced polyester resin gear case cover with T-stiffeners added after initial test failures.

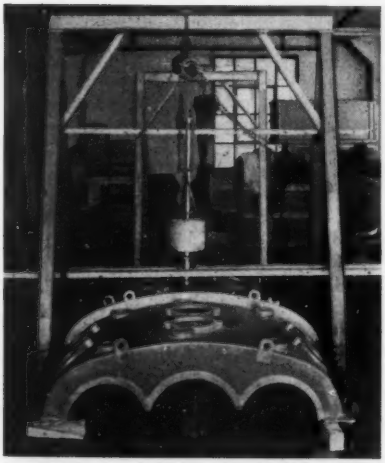

Figure 6. Single skin polyester resin cover, as initially manufactured, in laboratory impact test rig.

areas, entrapped air or porosity were not permitted.

The core materials were not to absorb more than 0.1 pounds of water or oil per sq. ft. of cut surfaces, when immersed in either, for 24 hours. Density was to be less thon 12 pounds per cu. ft.

Laboratory tests to determine impact strength included:

1—A drop of 2 feet of each half of each cover from any position. Such a drop was not to render the unit unfit for service.

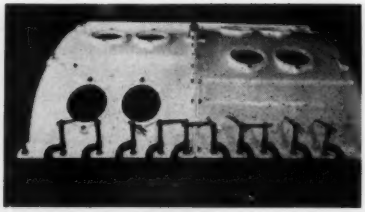

Figure 7. Single skin modified phenolic resin reinforced fiber-glass gear case cover.

2—Bolting the cover to a rigid frame Fig. 6, to simulate ship installation and subjecting it to a static load of 300 pounds distributed over a maximum area of 5 square inches without deflecting more than ¼ inch. In the location of greatest deflection the load was to remain in place for 12 hours without causing a permanent deflection greater than ⅛ inch.

3—The cover, while bolted in place as just described was to be subjected to an impact of a 300 pound weight dropped from a height of 2 feet. This impact load was to be distributed over a maximum area of 100 square inches and the case was not to deflect more than ¼ inch, or fail in any manner. The test apparatus is shown in Figs. 6 and 8.

MANUFACTURE AND FABRICATION

Three different types of materials and construction were used in the design and manufacture (all covers were manufactured by the E. L. Cournand and Co. Inc., Havre de Grace, Md.) (3) Each cover has the same overall dimensions (Table I,) and openings for inspection and thermometers. The materials (see Table II for structural properties), and fabrication are described briefly in the following paragraphs.

As previously mentioned, two covers were of single skin construction, and one was of the sandwich type. The reinforcements, Style 1000 fibre-glass cloth with Nr. 136 finish, was the same for all three units.

Single skin construction—Reinforced Polyester Resin

The resin used was a general purpose

rigid, styrene cross-linked, polyester resin. The method of fabrication was a hand layup of the main portion of the cover to a thickness of 3/16 inch, using style 1000 cloth with 136 finish. Separate press moldings were made for the access door bosses and temperature indicator opening bosses. These were then secondarily bonded to the cover as were the edge reinforcements. Threaded steel inserts were then installed in the bosses for cover attachments. A T-section stiffener was bonded to the inside surface of the cover, and two similar sections were added to the outside surface when load testing indicated their need. The cover when completed was coated on the outer surface with a fire retardant compound of antimony oxide and chlorowax.

Table II—Structural Properties

Results of Physical Tests on Sample Panels

	Polyester-glass single skin	Sandwich Constr. Polyester-glass, CCA Core	Modified Phenolic Resin
Flexural Strength—psi	26,500-30,700	20,000*	19,000
Flexural Modulus—psi	$1.54 \times 10^6 - 1.70 \times 10^6$	0.99×10^6*	1.10×10^6
Glass Content—percent	48.5	36.0*	55.5
Flammability	Self extinguishing	Self extinguishing*	burned completely
Oil Resistance	No effect	No effect	No effect
Water absorption—percent	0.14-0.27	6.8†	0.62

*Note 1—Properties determined for walls only, without CCA filler.
†Note 2—Water absorption was relatively high, but was caused by exposed CCA edges. No such exposed edges exist in gear case cover.

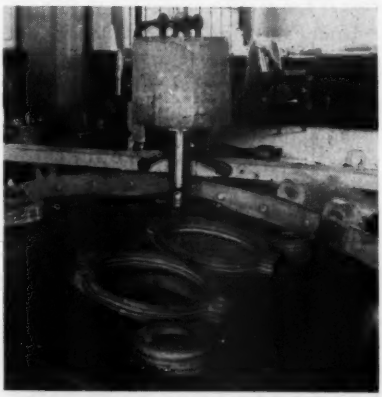

Figure 8. Gear case with 300 pound-weight used to apply impact load in laboratory test.

Sandwich Construction, Fig. 1

The second cover was a sandwich type construction employing style 1000-136 finish woven reinforcement and general purpose, polyester resin as the inner and outer skins with a 7 lb. per cu. ft. cellular cellulose acetate core.

The inner skin was first premolded on a male form. Then C.C.A. core was tailored to fit the cover contour and bonded to the inner skin with polyester resin as the adhesive. Following this, the outer surface was layed up by hand over the C.C.A. core. Skin thicknesses were ⅛" inner and 3/16" outer; the core was ⅝ inch thick, for an overall sandwich thickness of 15/16". The bosses and edge reinforce-

ments were bonded the same as in the cover described above. The unusual skin thickness was dictated by the severity of the impact test specified. Bosses for the access doors and temperature indicator openings were then bonded the same as those for the single skin type.

Plain Construction—Reinforced Phenolic Modified Resin, Fig. 7

The method for construction of this cover was the same as for the single skin cover. The only exception was that a modified room temperature curing phenolic resin was used instead of the polyester material.

LABORATORY TEST

A series of controlled tests (2) were conducted by the Materials Laboratory, New York Naval Shipyard on each of the covers. This testing was done in order to quickly eliminate covers which could be expected to fail under shipboard operation. The tests disclosed a number of design factors that were not acceptable. As a result of the laboratory work the polyester glass with single skin and the sandwich construction, polyester glass with CCA core, were selected as satisfactory. However, the selections were made only after some design changes were incorporated.

The question as to how the plastic covers would compare with a conventional steel cover under laboratory tests was answered by subjecting a steel cover to the same series of impact tests. Results of this test show that under impact, the maximum deflection of the steel cover as well as the plastic covers exceeded the specifications for the plastic covers, testifying to the severity of this test.

The results of laboratory tests on the various covers is summarized as follows:

Single skin, polyester-glass, Fig. 6

Under the 2 foot drop test several of the lifting tabs broke away from the main cover. During the impact test, with 300 pound weight, internal delamination occurred around several flanges, and deflection range was from 0.87 to 1.81 inches. The fire retardant properties were not satisfactory. This cover was redesigned, incorporating external T-shaped stiffeners and U-bolts in place of the plastic lifting tabs.

Single skin, polyester-glass, redesigned, Fig. 5

Under impact test one internal stiffener cracked and a slight delamination was noticed at the base of one flange. The cracked stiffener was replaced with a T-shaped one and no further tests were considered necessary. The cover was coated with a fire retardant of chlorowax and antimony oxide, and the lifting U-bolts replaced with eyebolts.

Sandwich Type, polyester glass, CCA core, Figs. 1, 2, 3, and 4

Under the impact test, slight cracking, and delamination occurred near the bosses. This was not considered serious and after being repaired by the manufacturer, no further testing was considered necessary. The cover was found to be self extinguishing. It was coated with chlorowax and antimony oxide by the manufacturer before delivery.

Maximum deflection of this cover under impact was 1.56 inches but no permanent deflection was encountered. Even though the deflection exceeded that specified it was considered acceptable due to the severity of the test and the fact that no permanent set was recorded.

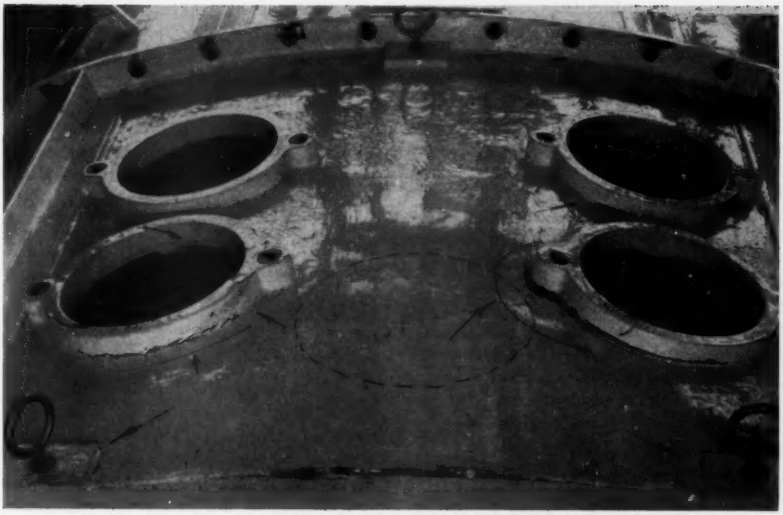

Figure 9. Single skin modified phenolic resin, reinforced fiber-glass cover after impact test. Dotted circle is impact area. Arrows point out delaminations and tears. A-19

Single Skin, modified phenolic resin, reinforced fibre glass, Fig. 7

Under the impact test, this cover showed low physical strength and deflection was as high as 2.50 inches. Delamination occurred around several bosses and some of the material pulled loose. In Fig. 9 the dotted circle is the impact area and the arrows point out the delaminations and tears. One of the internal stiffeners was partly delaminated and the flame proof coating chipped off. The cover was found to be not self extinguishing. This cover was considered unsuitable for naval use.

SHIPBOARD INSTALLATION

The sandwich construction cover Figs. 1, 2, 3, and 4 was selected for the initial ship installation. Prior to installation, the lifting attachments, which were U-bolts, with nuts on the inside of the cover, were replaced with eye bolts screwed into blind holes. This change was made to eliminate the possibility of a nut breaking away and falling into the gear mesh and also to reduce oil leakage.

Installation was made in several steps. First the plastic cover was placed on the gear case and all flange holes marked for drilling in the shop. The steel cover was then reinstalled and noise measurements taken to determine the airborne noise in the engine room that emanated from the reduction gear, while the ship was underway. Following the noise measurements the plastic cover was installed, Fig. 2, 3 and 4, and noise measurements repeated to determine any attenuation due to the plastic cover.

Some minor difficulties were experienced in the installation of the plastic cover. The location of thermometers proved critical. It was necessary to fit brass adapters, with a slight offset, to permit installation of thermometers in proper locations. The center transverse joint was fitted with a tapered spacer piece to bring the end flanges into alignment with the bearing housing flanges. Metal inserts were omitted where the steel end bells were attached to the plastic covers. The plastic was merely tapped and the studs screwed into place. Some oil leakage was encountered at the horizontal joint between the plastic cover and the steel case. This was corrected by installation of a gasket of sheet fiber (ND spec. 33-P-2) .005" to .007" and a sealing compound, Permatex No. 3 (aviation permatex).

Airborne noise measurements (4) were taken with steel cover in place and then with the plastic cover installed. These measurements were taken at six positions with pickups located 6 inches from the gear case at a propeller speed of 290 RPM. The plastic cover showed about 5 decibels average reduction in airborne noise and as much as 8 decibels at certain frequencies, Table III. This is not a startling reduction yet is is certainly a desirable one.

Table III—Noise Measurements
Measured Noise Levels on USS Fitch (DMS25)
Octave Band Level in Decibels

	Measured Decibels at 290 Propeller RPM	
Frequency Band in Cps.	Steel Cover	Plastic Cover
75-150	91	90
150-300	96	92.5
300-600	94.5	92
600-1200	101	95
1200-2400	100	92
2400-4800	93	86

SUGGESTIONS FOR NEW DESIGNS

As a result of laboratory testing of the three plastic covers, shipyard installation work and the limited shipboard experience the following points are offered for consideration of new design covers:

1—The design should be made to fit the material used and not necessarily duplicate the configuration of the steel cover.

2—Add reinforcing as needed to give strength with only secondary concern for appearance.

3—Install soft gasket, such as neoprene between flanges of plastic cover and steel case where oil tightness is necessary.

4—Incorporate metal reinforcing or use continuous metal strip with suitable bolt holes instead of single metal washers. Metal reinforcing or continuous strip washer will prevent bowing of flange between bolts.

5—Drill all bosses and flanges at the time of installation to assure accurate location of bolt holes, thermometer connections and inspection openings.

6—Leave outside diameter of bosses ½ inch oversize where practicable.

7—Design all flanges at least ⅛ inch thicker than corresponding steel parts.

8—Provide a plastic spacer for center transverse joint to allow for aligning of end flanges.

CONCLUSIONS

While the plastic cover has been installed on the USS *Fitch* a relatively short time, about six months, reports indicate it is performing in a satisfactory manner. It is yet too early to draw final conclusions; these will follow many months of operational evaluation, but based on the limited experience the plastic cover is satisfactory.

The single skin polyester-glass cover was also considered suitable for installation after laboratory testing. Present plans include installation of this cover on a second DMs to gain operational experience. If these covers prove satisfactory this type construction could be utilized in many future naval vessels with appreciable gains in weight reduction and possibly quieter operation.

Among the advantages of the plastic cover the following are outstanding:

1—*Weight reduction.* Up to 70% with a saving of 2,244 pounds per ship.

2—*Ease of handling.* Cover can be lifted or installed with less difficulty.

3—*Noise reduction.* The installed sandwich type cover reduces airborne noise radiating from reduction gear about 5 decibels average and up to 8 db. at certain frequencies. It is not expected as much noise reduction will be achieved when the single skin cover is installed.

4—*Non-critical materials.* The materials used in manufacture are considered non-critical.

5—*Production cost.* Quantity production should result in a cost that is about 50 percent of the steel cover.

6—*Non-corrosive.* No rusting or corroding as in a steel cover.

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THE FATHER OF AMERICAN NAVAL ENGINEERING

ACKNOWLEDGEMENT

Commander Neville T. Kirk, U.S.N.R., is the author of this article which was published in the April 1955 edition of *U. S. Naval Institute Proceedings*. Copyright, 1955, by the U. S. Naval Institute.

The train of events which followed the firing on Fort Sumter in the space of three months brought the United States Navy face to face with the most exacting task of its history to that time: the blockade of a 3500 mile coastline extending from Cape Hatteras to the Rio Grande. Of the 50 warships available for service only 28 had auxiliary steam power, and it soon became apparent that even these were no match for the elusive steamers on which the Confederates came increasingly to rely for their military supplies. The blockade eventually became effective in preventing the South from selling its cotton abroad to replenish its armies at home, but in order to accomplish this a new Navy had virtually to be created. One hundred and fifty warships of various types were built for the blockade and another 75 cruisers were obtained by converting merchant steamers. Still another navy of ironclads and "tinclads" was brought into being to support army movements on the inland waters. All these operations pointed up one fundamental fact: the steam engine which alone made them possible had by 1865 put sail as a prime motive power for navies into the same limbo as the oar. The symbol and, to a very considerable extent, the architect of the change was the Engineer-in-Chief of the Union Navy, Benjamin Franklin Isherwood. The occasion, in this instance, did indeed call forth the man.

Appointed to the Navy in 1844 as a Second Assistant Engineer Isherwood served in all the Gulf operations of the Mexican War. In the principal action, at Vera Cruz, he found himself in the hottest spot of all, in the engine room of Tatnall's *Spitfire* which lay unsupported under the guns of San Juan de Ulloa castle. He finished the war a Chief Engineer and was shortly afterward appointed technical assistant to Charles B. Stuart, the Engineer-in-Chief of the Navy. In a letter so cogently argued as to be included in the Secretary's Annual Report for 1851 Isherwood began the battle to put the service in the van of technical progress which he continued for the rest of his professional life. He pointed out that boiler power per unit weight had doubled in less than a decade, and he urged the re-boilering of the pioneer

National Archives Photograph

CHIEF ENGINEER BENJAMIN F. ISHERWOOD, U. S. NAVY

This hitherto unpublished Brady photograph was taken in Washington in 1864, at the height of the Civil War. As Chief of the Bureau of Steam Engineering with the relative rank of Commodore, Isherwood occupied a position of importance second to none in the direction of the first naval war which was won primarily by the use of steam power. His achievements in engineering science are summarized in the accompanying article.

Mississippi and her sister steam frigates even though their plants were, in the eyes of senior naval officers, relatively new. Another spell of sea duty in the *San Jacinto* which included active service at the Barrier Forts near Canton was followed, in March 1861, by Isherwood's appointment as Engineer-in-Chief.

Congress, responding to the emergency created by the unprecedented wartime expansion of the Navy, established the Bureau of Steam Engineering in July, 1862, and the infant grew so rapidly that at the end of hostilities it overshadowed all the other bureaus. Isherwood, its chief, putting in normal work days of 16 hours, was personally responsible for the design of main engines in a majority of the new ships, and in addition he handled the details of bureau administration and supervised the construction, installation and repair of all naval machinery.

In setting up a blockade on larger scale than had ever been attempted before, even by the "storm tossed ships" that brought Napoleon to his knees, the Navy Department was guided at every step by Isherwood's counsel. He suggested that an offshore line of heavy ships be stationed a hundred miles out and an inshore line cover the inlets and harbor mouths. He supplied designs for the machinery of the 20 *Hassalo, Contoocook,* and *Antietam* class cruisers which were laid down to man the outer line, though not all of these were completed. For shallow waters and the rivers his "double-ender" gunboat concept was adopted, and Bureau engineers under his direction supplied contractors with designs for the direct-acting paddle engines. But Isherwood made his distinctive contribution to practical naval engineering in the "*Alabama* chaser" *Wampanoag,* finished a couple of years after the close of the war.

The idea of this kind of fast cruiser originated with him. The Navy Department, however, took advantage of the opportunity presented by authorization to construct seven of the type to settle the bitter controversy over engine design which had flared since the first days of the war, and contracted with leading shipbuilders and engineers, including John Ericsson, to supply engines of their own design for the various ships in competition with the Bureau. In the steam trials that followed the *Wampanoag* won hands down. Isherwood, unlike his competitors, fully understood the need to avoid racking strains in wooden hulls. Accordingly, the *Wampanoag's* engines were fitted with gears which doubled the revolutions of the propeller shaft in relation to those of the crankshaft and thereby minimized the number of reciprocations of heavy engine parts. At 64 propeller revolutions the *Wampanoag* achieved a sustained sea speed of 17 knots, four more than any other warship afloat and a rate not attained by another American naval vessel for twenty years.

Meanwhile Isherwood was engaged in notable research. His two volumes of *Engineering Precedents* which appeared in 1859 represented the first systematic attempt of an American engineer to determine power losses through friction in a marine engine. At the height of the Civil War he published another two volumes of *Experimental Researches in Steam Engineering* which were quickly translated into six European languages. He was the first who had the courage to attack the classic theories of Watt and Marriott and to establish the point beyond which further expansion of the working charge in an engine cylinder produces net loss rather than a gain. And that point, he proved was earlier in the stroke than was believed by other engineers in that day of low working pressures. The fact not only created

astonishment but brought on Isherwood a storm of personal and professional abuse. But it also led directly to the adoption of higher steam pressures and the compound engine.

Isherwood inaugurated systematic engineering experimentation by the United States Navy. He was the first to show the proper way to test an engine and determine where the losses lay. To Rankine's deductions in thermodynamics relating to the "ideal case" he added the physical theory of thermal and dynamic losses which made the applied science of the heat engine of practical value to the engineer.

When peace settled over the land after Appomattox no threat to the national security appeared from abroad and the Navy experienced drastic retrenchment. The hard-driving Isherwood failed of re-appointment to the Bureau when his tour expired in 1869, and his floating powerhouse, the *Wampanoag,* was indefinitely laid up. In the interests of economy the General Order of June 18, 1869, forbade commanding officers to use coal except in extreme emergency, and added that they "must not be surprised, if they fail to carry out the spirit of this order, if the coal consumed is charged to their account." In justice it must be said that the marine engine was still a prodigious consumer of fuel. Some years later, in 1875, the British cruisers *Topaze* and *Doris,* buffeting monsoon headwinds in the Indian Ocean, reached Trincomaly with the former feeding furnaces with masts, spars, and the more weevily of her biscuits mixed with coaldust. The latter was stoking hemp cables. The marine engine was also highly unreliable; up to the early 1890's transatlantic liners of the largest size were on occasion left helpless at sea and unable to proceed to port except under tow.

Understandably it has come to be believed that American naval engineering went into a long coma during the quarter century after the Civil War. The facts, however, hardly bear out the assumption. Four years after the *Wampanoag's* triumphant performance Congress authorized construction of the three *Ranger* class iron cruisers, primarily powered by compound engines equal to the best foreign units of their size. A decade later, in 1883, the Bureau was engaged in designing what were then considered modern engines for the recently authorized cruiser *Chicago,* the largest ship of the new steel Navy. By 1893 stagnation in American naval construction was definitely over, and in that year the Bureau enjoyed the satisfaction of watching its engines drive the new *Minneapolis* more than 23 knots to a world's speed record for seagoing warships.

In these years Isherwood served on various boards but chiefly concentrated his energies on the investigation of propeller problems. Before his retirement in 1884 he compiled a body of data which served usefully to promote the later definitive propeller research of Admirals David W. Taylor and Charles W. Dyson. He was the acknowledged dean of the engineering profession in the United States when he died in his ninety-third year on June 19, 1915.

A worthy successor to the great American whose name he bore, Benjamin Franklin Isherwood served his country with distinction in science and in war. The relation of logistics to military victory has received proper recognition outside the military profession only in recent times. It is therefore regrettable but wholly explicable that historians have been less perceptive of Isherwood's achievements than a wartime subordinate, later head of the engineering school at Cornell University, who wrote: "It is, of course, true that he had at his command all the enormous resources of his bureau and the Navy Department, and could secure the aid of the ablest men of his

corps; but the organization and conduct of such enterprises, at such a time and under such extraordinarily difficult conditions . . . is evidence of the possession at once of moral, intellectual, and physical powers far beyond the average, and a spirit and ambition such as few men, even among the famous characters of historic times, have either possessed or concentrated upon any great purpose."

FORMULAS FOR THE DRIVING POINT ADMITTANCE OF FOLDED WIRE STRUCTURES

COMMANDER CHARLES W. HARRISON, JR., U.S.N.

THE AUTHOR

attended the U. S. Naval Academy Preparatory School, the U. S. Coast Guard Academy and the University of Virginia, where in 1939 he received the S.B. degree in Engineering and in 1940 the degree of Electrical Engineer. In 1942 he was graduated with the S.M. degree in Communication Engineering from Cruft Laboratory, Harvard University, and that summer completed the Navy course in Radar Engineering at Massachusetts Institute of Technology. Subsequently for several years he was engaged in lecturing to officers of the Armed Forces assigned to the radar schools at Harvard and Princeton Universities. He has had four tours of duty in the Electronics Design and Development Division, Bureau of Ships; two at the U. S. Naval Research Laboratory; one at the Signal Corps Engineering Laboratories (Evans Signal Laboratory); and one at the Philadelphia Naval Shipyard. His experience includes amateur, naval and broadcasting operation.

In 1951 Commander Harrison was selected for training in Advanced Science at Harvard University under sponsorship of the Office of Naval Research. This program led to the M.Eng. degree in 1952 and Ph.D. degree in Applied Physics in 1954.

Commander Harrison is presently Electronics Officers on the Staff of Commander Operational Development Force. He is a member of several societies and associations, including the Harvard Chapter of the Society of Sigma Xi.

SUMMARY: Formulas are presented, without derivation, for the driving point admittance of twenty-six folded wire configurations. These formulas, together with ancillary tables recently published, constitute a "folded antenna handbook." It is believed that these data will prove useful to engineers engaged in antenna design.

INTRODUCTION

The purpose of this paper is to present a compilation of formulas deduced by the writer for the driving point admittance of various folded wire structures. These formulas, together with the "Universal Tables for Folded Antennas" recently published [1] permit rapid numerical studies of the admittance qualities of

folded wire configurations as influenced by changing the dimensions and spacing of the circuit elements.

The half-length of each center-driven structure is h. The cylindrical conductors, which are assumed to be perfectly conducting, are of radius a (or a_1 and a_2 if a distinction is necessary). The spacing between adjacent wires is b. For valid results the conditions $a<<h$, $\beta a<<1$ and $\beta b<<1$ must be satisfied. Here β is 2π divided by the wavelength. Further, the folded antenna-transmission line must be isolated from its surroundings. This condition is satisfied if $\beta R>>1$ and $R>>h$. R is the distance from the center of the structure to the nearest point on any conductor or dielectric.

If the number of conductors comprising a circuit exceeds 2 it is presumed, unless otherwise stated, that the cross-section of the structure, taken at right angles to the wire axes, is that of a regular polygon; i.e., equilateral triangle, square, regular pentagon, etc.

The driving point admittance is designated Y_{in}; the driving point impedance of the "comparable" symmetrical center-driven antenna Z_d and the "effective" radius d.

To determine Z_d as a function of β h from the universal tables [2] necessitates knowing the value of Ω. The formula for this paremeter is $\Omega_d=2\log\frac{2h}{d}$.

Throughout this article $\log x \equiv \ln x$.

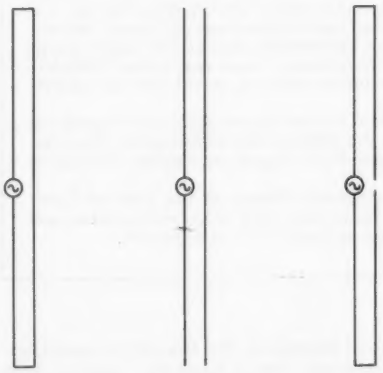

Fig. 1 Fig. 2 Fig. 3

PART I

Folded Wire Structures Consisting of Wires of the Same Radius

1. Conventional Two-Wire Folded Dipols. Figure 1.

$$Y_{in}=\frac{1}{4Z_d}-j\frac{\cot\beta h}{240\log\frac{b}{a}}\text{mhos}$$

$$d=\sqrt{ab}$$

2. Two-Wire Structure with Open-Circuited Ends. Figure 2.

$$Y_{in}=\frac{1}{4Z_d}+j\frac{\tan\beta h}{240\log\frac{b}{a}}\text{mhos}$$

$$d=\sqrt{ab}$$

3. Two-Wire Folded Antenna-Transmission Line with Air Gap. Figure 3.

$$\frac{1}{Y_{in}} = Z_d + j\,60 \log \frac{b}{a} \tan \beta h \text{ ohms}$$

$$d = \sqrt{ab}$$

Figure 4

Figure 5

4. Symmetrical Center-Driven Antenna in Proximity to an Open-Circuited Parasitic Element. Figure 4.

$$\frac{1}{Y_{in}} = Z_d - j\,60 \log \frac{b}{a} \cot \beta h \text{ ohms}$$

$$d = \sqrt{ab}$$

5. U Antenna-Transmission Line. Figure 5.

$$Y_{in} = \frac{1}{4Z_d} + j \frac{\tan 2\beta h}{240 \log \frac{b}{a}} \text{ mhos}$$

$$d = \sqrt{ab}$$

Figure 6

Figure 7

Figure 8

6. Three Conductor Folded Antenna having Two Open-Circuited Ends. Figure 6.

$$Y_{in} = \frac{1}{9Z_d} - j\,\frac{1}{90 \log \frac{b}{a}} \left\{ \frac{\sin 2\beta h}{1 - 2\cos 2\beta h} \right\} \text{mhos}$$

$$d = \sqrt[3]{ab^2}$$

7. Conventional Three-Wire Folded Dipole. Figure 7.

$$Y_{in} = \frac{1}{9Z_d} - j\,\frac{\cot \beta h}{180 \log \frac{b}{a}} \text{mhos}$$

$$d = \sqrt[3]{ab^2}$$

8. Two-Wire Folded Dipole with Single Conductor Branch. Figure 8.

$$Y_{in} = \frac{1}{9Z_d} - j\,\frac{\cot \beta h (3 - 8\sin^2 \beta h)}{720 \log \frac{b}{a} (\cos^2 \beta h - \sin^2 \beta h)} \text{mhos}$$

$$d = \sqrt[3]{ab^2}$$

Figure 9

Figure 10

9. Symmetrical Center-Driven Antenna in Proximity to a U Shaped Parasitic Element.

$$Y_{in} = \frac{1}{9Z_d} + j\frac{\tan \beta h}{180 \log \frac{b}{a}} \text{ mhos}$$

$$d = \sqrt[3]{ab^2}$$

10. U Antenna in Proximity to a Short-Circuited Parasitic Element. Figure 10.

$$Y_{in} = \frac{1}{9Z_d} + j\frac{\tan \beta h (8 \cos^2 \beta h - 1)}{720 \log \frac{b}{a} (\cos^2 \beta h - \sin^2 \beta h)} \text{ mhos}$$

$$d = \sqrt[3]{ab^2}$$

Figure 11

Figure 12

11. Symmetrical Center-Driven Antenna in Proximity to a Short-Circuited Turn of Wire. Figure 11.

$$Y_{in} = \frac{1}{9Z_d} + j\frac{\tan \beta h}{180 \log \frac{b}{a}} \text{ mhos}$$

$$d = \sqrt[3]{ab^2}$$

12. Conventional Two-Wire Folded Dipole in Proximity to a Short-Circuited Parasitic Element. Figure 12.

$$Y_{in} = \frac{1}{9Z_d} - j\frac{\cos 2\beta h + 2\cos^2 \beta h}{360 \log \frac{b}{a} \sin 2\beta h} \text{ mhos}$$

$$d = \sqrt[3]{ab^2}$$

Figure 13

Figure 14

13. Double U Antenna-Transmission Line. Figure 13.

$$Y_{in} = \frac{1}{9Z_d} + j\frac{\tan 2\beta h}{180 \log \frac{b}{a}} \text{ mhos}$$

$$d = \sqrt[3]{ab^2}$$

14. Three-Wire Asymmetrical Folded Antenna Having Two Open-Circuited Ends. Figure 14.

$$Y_{in} = \frac{1}{9Z_d} - j\frac{\sin \beta h \cos \beta h \ (24 \sin^2 \beta h - 9)}{90 \log \frac{b}{a} (9 - 12 \sin^2 \beta h)(1 - 4 \sin^2 \beta h)} \text{ mhos}$$

$$d = \sqrt[3]{ab^2}$$

Figure 15

Figure 16

15. Single Conductor Antenna with Shorted-Turn Branch. Figure 15.

$$Y_{in} = \frac{1}{9Z_d} + j\frac{\tan 2\beta h}{180 \log \frac{b}{a}} \text{ mhos}$$

$$d = \sqrt[3]{ab^2}$$

16. Three Conductor Antenna-Transmission Line having Center as well as End Folds. Figure 16.

$$\frac{1}{Y_{in}} = Z_d + j\frac{160 \log \frac{b}{a}}{\cot \beta h + \cot 2\beta h} \text{ ohms}$$

$$d = \sqrt[3]{ab^2}$$

Figure 17

Figure 18

17. Conventional Four-Wire Folded Dipole. Figure 17.

$$Y_{in}=\frac{1}{16Z_d}-j\,\frac{\cot\beta h\,\log\frac{b^6}{2a^6}}{240\log\frac{2b^2}{a^2}\log\frac{b^2}{2a^2}}\text{mhos}$$

$$d=\sqrt[4]{ab^3\sqrt{2}}$$

18. Four-Conductor Antenna-Transmission Line having Center as well as End Folds. Figure 18.

$$Y_{in}=\frac{j}{240\log\frac{2b^2}{a^2}}\left\{\tan\beta h-\frac{\cot\beta h}{2\log\frac{b}{a\sqrt{2}}}\left(\log\frac{2b^2}{a^2}\left[1\right.\right.\right.$$

$$\left.\left.\left.-2\,\frac{8\pi Z_d\cot\beta h-j\,240\pi\log\frac{b}{a\sqrt{2}}}{8\pi Z_d\cot\beta h+j\,240\pi\log\frac{b}{a\sqrt{2}}}\right]+4\log\frac{b}{a}\right)\right\}\text{mhos}$$

$$d=\sqrt[4]{ab^3\sqrt{2}}$$

Figure 19

Figure 20

Figure 21

19. Four-Conductor Re-entrant Loop. Figure 19.

$$Y_{in} = \frac{1}{16Z_d} - j\,\frac{\cot \beta h \left(2 \sin^2 2\beta h \log \frac{2a^6}{b^6} + \log \frac{2b^2}{a^2}\right)}{240 \log \frac{2b^2}{a^2} \log \frac{b^2}{2a^2} \cos 2\beta h} \text{ mhos}$$

$$d = \sqrt[4]{ab^3\sqrt{2}}$$

20. Conventional Five-Wire Folded Dipole. Figure 20.

$$Y_{in} = \frac{1}{25Z_d} - j\,\frac{\cot \beta h \log \frac{1.618b^2}{a^2}}{300\left[\log \frac{b}{a} \log \frac{1.618b}{a} - 0.2315\right]} \text{ mhos}$$

$$d = \sqrt[5]{2.618\,ab^4}$$

21. Five-Wire Symmetrically Folded Antenna having Two Open-Circuited Ends. Figure 21.

$$Y_{in} = \frac{1}{25Z_d} - j\,\frac{\cot \beta h \left\{5 \log \frac{b}{a} + \log \frac{a^3}{6.854b^3} \cot^2 \beta h\right\}}{75 \left\{\log \frac{b}{a} \log \frac{1.618b}{a} - 0.2315\right\} \left\{5 - 10 \cot^2 \beta h + \cot^4 \beta h\right\}}$$

$$d = \sqrt[5]{2.618\,ab^4}$$

PART II

Folded Wire Structures Consisting of Wires of Unequal Diameter

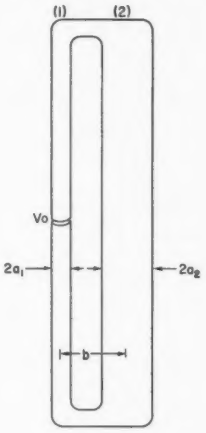

Figure 22

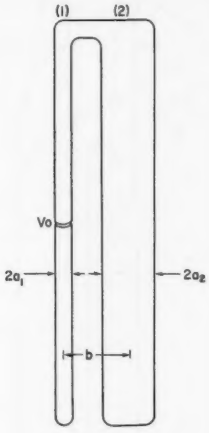

Figure 23

1. Conventional Folded Dipole Consisting of Two Wires of Unequal Diameter. Figure 22.

$$Y_{in} = \frac{1}{Z_d}\left(\frac{\log\frac{b}{a_2}}{\log\frac{b^2}{a_1a_2}}\right)^2 - j\,\frac{\cos\beta h + 2\Gamma\log\frac{a_1}{a_2}}{120\log\frac{b^2}{a_1a_2}\sin\beta h}\ \text{mhos}$$

$$d = \sqrt{a_1 b}\cdot e^{-\frac{\log\frac{b}{a_1}\log\frac{a_1}{a_2}}{2\log\frac{b^2}{a_1a_2}}}$$

Γ is definied in Appendix I. The ratio V_K/V_0 occurring in the formula for Γ is $V_K/V_0 = \log\frac{b}{a_2} \Big/ \log\frac{b^2}{a_1a_2}$. The wire of radius a_1 is centerdriven.

2. U Antenna-Transmission Line Consisting of Wires of Unequal Diameter. Figure 23.

$$Y_{in} = \frac{1}{Z_d}\left(\frac{\log\frac{b}{a_2}}{\log\frac{b^2}{a_1a_2}}\right)^2 + j\,\frac{\sin\beta h(\cos\beta h + \Gamma\log\frac{a_2}{a_1}}{60\log\frac{b^2}{a_1a_2}\cos 2\beta h}\ \text{mhos}$$

d and Γ have the same values as in Part II, 1.

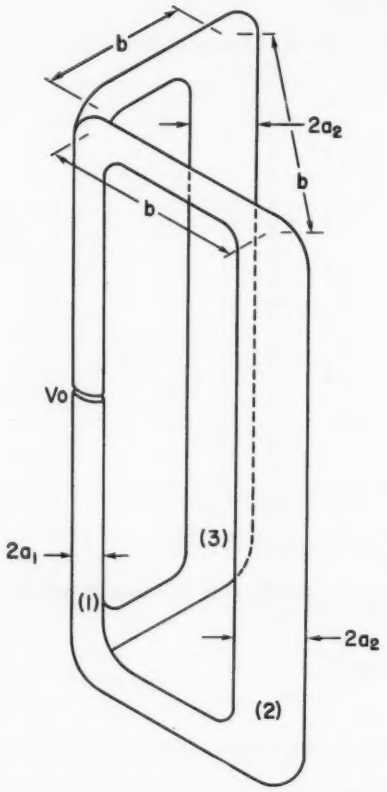

Figure 24

3. Conventional Three-Conductor Folded Dipole Consctructed of Two Sizes of Wire. Figure 24.

$$Y_{in} = \frac{1}{Z_d}\left(\frac{\log\frac{b}{a_2}}{\log\frac{b^3}{a_1 a_2}}\right)^2 - j\,\frac{1.5\cos\beta h + 2\Gamma\log\frac{a_1}{a_2}}{90\log\frac{b^3}{a_1^2 a_2}\sin\beta h}\ \text{mhos}$$

$$- \frac{\log\frac{b}{a_1}\log\frac{a_1^2}{a_2 b}}{2\log\frac{b^3}{a_1^2 a_2}}$$

$$d = \sqrt{a_1 b} \quad e$$

$$V_K/V_0 = \log\frac{b}{a_2} \Big/ \log\frac{b^3}{a_1^2 a_2}$$

The axes of the wires lie on the vertices of an equilateral triangle. The wire of radius a_1 is center-driven. The other conductors are of radius a_2.

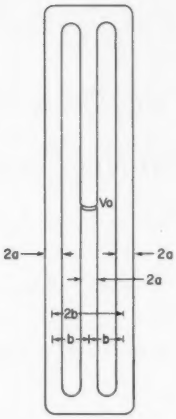

Figure 25

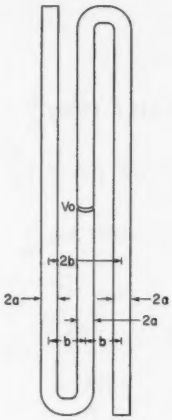

Figure 26

PART III

Folded Wire Structure Consisting of Wires of Equal Diameter Lying in a Plane

1. Conventional Three-Wire Folded Dipole. Figure 25.

$$Y_{in}=\frac{1}{Z_d}\left(\frac{\log\frac{b}{2a}}{\log\frac{b^3}{2a^3}}\right)^2-j\frac{\frac{1}{4}\cos\beta h-\frac{1}{3}\Gamma\log 2}{15\log\frac{b^3}{2a^3}\sin\beta h}\ \text{mhos}$$

$$d=\sqrt[4]{2ab^3}\ e^{-\frac{\log\frac{b}{2a}\log\frac{2b}{a}}{4\log\frac{b^3}{2a^3}}}$$

$V_K/V_O=\log\frac{b}{2a}\Big/\log\frac{b^3}{2a^3}$. The middle wire is center driven.

2. Three-Wire Folded Antenna with Two Open-Circuited Ends. Figure 26.

$$Y_{in}=\frac{1}{3\log\frac{b}{a\sqrt[3]{2}}}\left\{\frac{\left(\frac{1}{3}-\frac{0.154}{\log\frac{b}{a\sqrt[3]{2}}}\right)\log\frac{b}{2a}}{Z_D}-j\frac{K}{60}\right\}\ \text{mhos}$$

$$K = \sin \beta h \left\{ \frac{5.544\,\Gamma\,(4 \log \frac{b}{a} - \log \frac{2b}{a}) + \cos \beta h\,(6 \log \frac{2b}{a} - 30 \log \frac{b}{a} + 1.386)}{\cos^2 \beta h\,(6 \log \frac{b}{a} - 1.386) + \sin^2 \beta h\,(6 \log \frac{2b}{a} - 24 \log \frac{b}{a})} \right\}$$

$$d = \sqrt[3]{2ab^2}\ e^{-\frac{0.077 \log \frac{b}{2a}}{\log \frac{b}{a\sqrt[3]{2}}}}$$

$$V_K/V_0 = \left\{ \frac{1}{3} - \frac{0.154}{\log \frac{b}{a\sqrt[3]{2}}} \right\}$$

The middle wire is center driven.

APPENDIX

The complex parameter Γ is given by the formula

$$\Gamma = \frac{V_K}{2V_0 \psi_d} \left\{ \frac{1 - \frac{1}{\psi_d}\left(2 \sinh^{-1} \frac{h}{d} - \sinh^{-1} \frac{2h}{d} - \psi_d - 1\right)}{\cos \beta h + \frac{1}{\psi_d} F_1(h)} \right\}$$

Here

$$F_1(h) = \frac{1}{2} \cos \beta h \left\{ \overline{Ci}\ 4\beta h + j\,Si\ 4\beta h \right\} - \frac{1}{2} \sin \beta h \left\{ Si\ 4\beta h - j\,\overline{Ci}\ 4\beta h \right\}$$

$$- \cos \beta h \left\{ \overline{Ci}\ 2\beta h + j\,Si\ 2\beta h \right\}$$

$$\psi_d(z) = \frac{C_d(z) \sin \beta h - S_d(z) \cos \beta h}{\sin \beta (h - /z/)}$$

$$\psi_d = /\psi_d(0)/ \text{ if } \beta h \leq \frac{\pi}{2}, \text{ and } \psi_d = /\psi_d\,(h - \frac{\lambda}{4})/ \text{ if } \beta h \geq \frac{\pi}{2}.$$

λ is the wavelength.

$$C_d(z) = -\frac{1}{2}\cos\beta z \left\{\overline{Ci}\ 2\beta\,(h+z) + \overline{Ci}\ 2\beta\,(h-z)\right.$$

$$\left. + j\,Si\,2\beta\,(h+z) + j\,Si\,2\beta\,(h-z)\right\}$$

$$+\frac{1}{2}\sin\beta z \left\{Si\ 2\beta\,(h+z)\ -Si\ 2\beta\,(h-z)\right.$$

$$\left. - j\,\overline{Ci}\,2\beta\,(h+z) + j\,\overline{Ci}\,2\beta\,(h-z)\right\}$$

$$+\cos\beta z \left\{\sinh^{-1}\left(\frac{h+z}{d}\right) + \sinh^{-1}\left(\frac{h-z}{d}\right)\right\}$$

$$S_d(z) = \frac{1}{2}\cos\beta z \left\{Si\,2\beta\,(h+z) + Si\ 2\beta\,(h-z) - 2\,Si\,2\beta z - j\,\overline{Ci}\,2\beta\,(h+z)\right.$$

$$\left. - j\ \overline{Ci}\ 2\beta\,(h-z) + j\,2\ \overline{Ci}\,2\beta z\right\}$$

$$+\frac{1}{2}\sin\beta z \left\{\overline{Ci}\ 2\beta(h+z) - \overline{Ci}\ 2\beta\,(h-z)\right.$$

$$\left. - 2\ \overline{Ci}\ 2\beta z + j\ Si\ 2\beta\,(h+z) - j\ Si\,2\beta\,(h-z) - j\,2\ Si\ 2\beta z\right\}$$

$$-\sin\beta z \left\{\sinh^{-1}\left(\frac{h+z}{d}\right) - \sinh^{-1}\left(\frac{h-z}{d}\right) - 2\sinh^{-1}\left(\frac{z}{d}\right)\right\}$$

$$\overline{Ci}(x) = 0.5772 \qquad \log x - \overline{Ci}(x) = \int_0^x \frac{1-\cos u}{u}\,du$$

$$\overline{Ci}(x) = \int_\infty^x \frac{\cos u}{u}\,du \text{ (See reference 3, below)}$$

$$Si(x) = \int_0^x \frac{\sin u}{u}\,du \text{ (See reference 3, below)}$$

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[1] Charles W. Harrison, Jr., "Driving Point Admittance of a Symmetrically Folded Antenna," Journal of the American Society of Naval Engineers, Vol. 67, No. 1, pp. 213-238, February, 1955.

[2] Reference 1, pp. 225-238.

[3] Values of Ci(x) and Si(x) are tabulated in two volumes entitled "Tables of Sine, Cosine and Exponential Integrals" prepared by the Federal Works Agency, Work Projects Administration for the City of New York (1940). An extension of the same work appeared in 1942 entitled "Table of Sine and Cosine Integrals for Arguments from 10 to 100."

A NEW DESIGN FOR SMALL MOTORS

ACKNOWLEDGEMENT

Charles F. Irvin, of the Small Motor Division, Westinghouse Electric Corporation, Lima, Ohio, is the author of this article which was published in the March 1955 issue of *Westinghouse Engineer*.

Fractional-horsepower motors, like their integral-horsepower counterparts, are undergoing a general rerating program designed to produce a motor both smaller and lighter for a given power output. The new standard proposed by NEMA is the 48-frame size, which differs from the previous 56 frame primarily in shaft height (reduced from 3½ to 3 inches) and in mounting dimensions. However, in addition to reduced size and weight, the new Westinghouse fractional-horsepower motor has better insulation, a new ventilation system designed for efficient cooling at a low noise level, a unique new connection board, and numerous other design improvements.

The initial redesign to a 48-frame size encompasses Westinghouse motors in the four-pole, open, drip-proof, fully guarded ratings from one-third through one-eighth horsepower, continuous duty, 40 degree C temperature rise. Later designs will include the two-pole ratings up through one-half horsepower and the six-pole ratings up through the one-quarter-horsepower size.

Figure 1. A comparison in size between motors of 1903, 1946, and 1955; all are of ¼-hp, 1725-rpm, 60-cycle rating.

Actually, the present re-rating program is but part of the continuing trend toward smaller motors that are at the same time better electrically and mechanically. Contrast, for example, the one-quarter horsepower motor of 1903 with the new fractional-horsepower motor of the same rating (Fig. 1). The weight has been reduced from 69 pounds to but 16 for the new motor. The new motor is six pounds lighter than the motor of nine years ago—about a 27-percent reduction. Obviously, such a drastic cut in size and weight necessitated the solution of a number of engineering design problems.

Ventilation—A reduction in motor size without a decrease in power output makes an improvement in ventilation efficiency imperative. Since sound level is an important factor to the user, this increase in ventilation efficiency cannot be accompanied by greater windage noise—preferably there should be less. But in designing the new motor, engineers also gave themselves certain other restrictions; namely, that the ventilation system should produce the same amount of cooling regardless of the direction of motor rotation, and that all ventilation openings should be in the motor end shields so that universality of motor mounting would not be restricted by the ventilation system.

The final ventilation system developed is actually simple, but effective; it is based on obtaining maximum operating efficiency from the internal ventilating fan. A shrouded fan is mounted on the motor shaft; this fan draws cool air in at its center, through the rear end shield, and produces a positive-pressure, straight-through cooling. Intake air enters the fan perpendicular to the fan blades and at a maximum relative velocity for most efficient operation. Because it moves cold air, which is more dense than hot air, the fan moves a greater weight of air; the result is a 10-percent increase in ventilation ability over a system that displaces hot air.

The fan exhaust is an enclosed area inside the rear end shield of the motor. Air is diffused around the the motor coil extensions, passes over the rotor end ring into the rotor-core ventilation holds, and then over the coil extensions and out the front end shield (Fig. 2).

This system virtually eliminates the normally encountered windage noise at the fan-exhaust area, since the air is completely diffused and traveling at a relatively low velocity through the end-shield openings. Also, there are no revolving fan blades passing close to ventilation holes to cause a high-frequency whistle, as is characteristic with many designs. Also, the area of maximum air turbulence inside the rear end shield is entirely enclosed; the inside surface of the shield itself is purposely made extra smooth to prevent high-frequency windage noise. The result is a low-noise, high-volume ventilation system.

The cooling improvement over the previous (56-frame) motor is sizable.

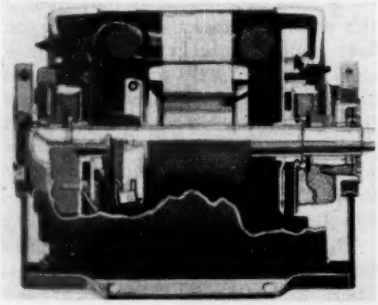

Figure 2. Cutaway of the new motor, showing path of ventilating air.

The hot-spot winding temperature of the one-quarter horsepower, open split-phase motor, for example, is 25 percent less than for the older motor. For the one-sixth horsepower motor the corresponding decrease is 40 percent, and for the one-third horsepower motor 10 percent.

Equally important is the fact that this new ventilation system affords cooler running bearings. The heavy load bearing at the shaft-extension end of the motor has a temperature rise of but 18 degrees C, compared to 35 degrees for the 56-frame motor. Lower operating temperature retards oxidation and evaporation rates of lubricants; since these are primary factors in the life of the lubricant, the useful life of the bearing is significantly increased.

Motor Protection—The straight-through ventilation system is tailored for the optimum protection afforded by the Thermoguard protective system. With straight-through ventilation, the thermostat is more responsive to abnormal temperature regardless of which end of the motor it occurs in.

Connection Board—The connection board for the motor incorporates in a single unit all the necessary connections. The new board is mounted in the wound primary of the motor on four support studs; this provides a rigid and accurately controlled means of maintaining the relative location of the connection board in the wound primary, so that solid wires from the winding coils can be attached directly to the connection board without special provision for flexing.

This concept of a unitized connection board has several advantages. There are absolutely no electrical connections inside the winding. This eliminates trouble caused by short circuits, grounds, and open circuits in the winding connections. Further, the assembly and disassembly of the motor is simplified by eliminating the flexible leads normally used for connections from the wound primary to the motor end shield.

A novel method for making rotation and voltage connection changes has been incorporated in the motor. Quick-disconnect plugs are used. Interchange of plugs in making winding connections requires no tools and line connections in the motor need not be disturbed. Change in direction of rotation of the motor is accomplished by the simple interchange of two quick-disconnect plugs. Additional connectors are provided on dual-voltage motors to allow switching from parallel to series connection, for changing from 115-volt to 230-volt operation. This takes a matter of perhaps ten seconds after the conduit-box cover is removed, compared to the three or four minutes that are required to change connections by other methods.

Additional Improvements—By a centerless grinding process, the complete shaft of the motor is machined at one time, which assures a precision finish on all diameters and a very concentric relationship between these diameters. All components utilized in the rotor core are carefully correlated to obtain a finished unit with the least possible electrical unbalance. Extreme emphasis on the control of concentricity of all parts going into the assembly of the motor assures excellent balance and low slip-frequency noise.

A new lubrication system affords a more than adequate supply of oil (some 22 percent more than the present 56-frame motor) and assurance that the oil will not leak out of the motor in any position, regardless of whether the motor is running or

standing still. The system is designed to give long years of service under loaded operating conditions.

The resilient base for the new motor is fabricated, which results in a 15-percent reduction in weight, with increased rigidity and strength.

Stator slots are insulated with a laminated combination of Mylar film and 100-percent rag paper bonded together. The combination has a high dielectric strength as well as high mechanical strength. The present rag-paper insulation has a total dielectric strength of 4500 volts, compared to 8500 volts for the new insulation—in spite of reduced thickness.

After the steel frame is applied to the wound core, the complete assembly is dipped in a thermosetting phenolic - resin insulating varnish, which gives a highly moisture-resistant coating over the whole wound primary and frame.

The end result is a motor that is not only physically smaller, but electrically and mechanically much superior to its larger predecessor. It represents one more step in the continuing effort to improve fractional-horsepower motors.

DIESEL WINGS IN THE NAVY'S FUTURE

ACKNOWLEDGEMENT

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Following the first encroachments on the sonic barrier, the exploits of aircraft such as the Douglas Skyrocket strained the public imagination to the breaking point. If two thousand miles per hour were possible this year, would not four thousand be reached next, and so on *ad infinitum*? The citizen's mind is now filled with dreams of automatically controlled nuclear aircraft, not to mention jet-borne vacations—New York to Paris in an hour or less.

These projections are understandable. Speed is apparently becoming the foremost criterion of air efficiency, and the "leading edge" of aircraft design depicts craft shaped like shells with skins of glass or ceramic construction underlaid with elaborate cooling systems to push them further through the thermal barrier.

Naval aviation in the United States has made commendable strides since the close of World War II. For a time, the Navy made the quest for the world's speed record seem like a private tussle with the U. S. Air Force. Developments in seadrome-based fighters, vertical take-off aircraft, and heavier and faster planes to take off from and land on carriers have caught up the civilian and the military man alike and carried them in the wake of bigger, faster, and more devastating aircraft.

Silently accompanying the display of speed engineering now in progress, however, lies the cold military truth that equipment designed for a specific task must accomplish that mission superbly, with only secondary consideration for secondary purposes. To hunt and kill a submarine maneuvering at 12 knots, one would not choose an aircraft cruising at 1,200 knots. The aircraft designed for speed is not at its best in slow flight. The logical choice is a craft which can emulate the submarine and pinpoint its movements. Hence, it is reasonable to assume that the patrol bombers of the future will have much in common with the lumbering PBY. The goal of flying faster, higher, and farther may well be left to the fighter and high-altitude bomber. Patrol craft, observation planes, low-level bombers, and transports are dependent on maneuverability, endurance, pay load, long range, and economy of operation—the very features which are so often shuffled to the background today.

The propeller-driven plane, which presents so many difficulties in adaptation to flight beyond the sonic barrier, has proved the ideal power plant

for patrol, reconnaissance, and transport work in the past, and will continue to hold sway in these fields for a number of years. Even in the race for speed, the propeller is still present. Turbo-prop combinations have met with great success and may prove the ideal balance between speed and economy.

At the mention of propeller driven aircraft, an air-cooled radial gasoline engine comes to mind, the engine which has pushed or pulled naval aviation through the air for the past thirty years. The gasoline engine is a remarkable device, but no more so than the compression-ignition, or diesel, engine which few people associate with aircraft. Since the end of World War II, the development of the diesel engine for aircraft use has taken such strides that the United States can no longer overlook the fertile possibilities of diesel aviation.

To make any comparison between the compression-and spark-ignition engines, one should begin with the inherent advantages of each type to the desired application, and not with the progress of gasoline-powered flight and the lack of progress in aircraft diesels. Because it must withstand compression and combustion pressures from 800 to 3,000 pounds per spare inch, the diesel is a heavier engine than the gasoline power plant. Offsetting this factor is the greater thermal efficiency possible with the diesel—38% for the gasoline engine and 45% with diesel power. Diesel fuel is cheaper, and the engine is more adaptable to different grades and types of fuel, but its complex fuel injection and compressor systems are noted for poor performance when the engine is subjected to varying altitudes. Once started, the diesel shows greater endurance and is less subject to accidental fires, but it is inherently hard to start. A compression-ignition engine will continue to operate after much greater structural damage but cannot match the spark-ignition engine in the high crankshaft speeds necessary for very rapid acceleration.

It was previously mentioned that there were four or five types of military aircraft now dependent on the propeller for motive force. Of these, the patrol bomber and the reconnaissance aircraft are what may be termed long-range propeller fields. It is doubtful that any other power plant will replace the internal combustion engine in these craft as long as warfare remains within the present bounds of our imagination. Low altitude bombers and freight and personnel transport planes may be termed short-range propeller fields. A drastic advance in jet, rocket, or nuclear propulsion may make propeller craft of these types obsolescent, but this is not likely in the immediate future.

It would be worthwhile to examine the primary missions of the types mentioned in order to understand the advantages which would be presented if aircraft diesel engines were made practical for these aircraft. The patrol bomber must successfully execute search, long range, convoy, and scouting patrols, antisubmarine warfare, bombing and torpedoing, transport, and rescue work. The keynote in patrol plane operation is range, endurance, payload, and durability. Scout, observation, and associated aircraft must accomplish missions of scouting and search, antisubmarine patrol, observation of gunfire, attack on light forces, and occasional photographic missions. These craft must depend on variable range, maneuverability, adaptability to weather and fuel conditions, and smooth operation over a wide speed

range. Transport aircraft must render full tactical and logistic support, whether paratroops or peanuts are needed. Safety and endurance are primary consideration; economy of operation and payload are also of prime importance. Low-altitude bombers differ little from World War II planes of this type. Their primary requisites are still moderate speed, good range, and ability to absorb punishment without failure. With the present plans for carrier complements of these aircraft, ease of control and maneuverability should be given adequate design consideration.

In both the long- and short-range propeller fields, it is evident that the aircraft diesel is an ideal power plant if four major difficulties could be surmounted:

1. High specific weight, the ratio of engine weight to engine brake horsepower.
2. Unreliability of fuel injection and compressor systems.
3. Difficulty in starting, and length of warm-up period.
4. Difficulty of sufficient and uniform engine cooling.

These hurdles are complicated by the fact that the two-cycle diesel is much better adapted to aircraft than the bulkier four-cycle engine. Two-cycle operation adds to fuel injection and cooling problems since the number of power strokes and number of injections per cylinder are doubled at equal crankshaft speed. The solution to these problems is nearer than it has ever been. It is now so near that the United States and the Navy in particular will find themselves at a disadvantage if the promise of aircraft diesel power is not diligently pursued.

The use of diesel power plants for aircraft is not a recent innovation. The diesel's promise of higher thermal efficiency and remarkable durability under conditions of marginal maintenance set off sporadic flurries among aircraft designers as early as 1913. In Germany Dr. Hugo Junkers developed a two cycle, four cylinder, opposed piston engine designated the MO-3. It was a successful aircraft engine in all respects except that it was never flown in a successful aircraft. The MO-3 was a remarkable power unit, developing 350 HP at a time when the most powerful gasoline engines were producing 300-325 HP. Interest in the early aircraft diesels had an economic, rather than an engineering impetus. The best gasoline engines of the day had a thermal efficiency of 25%, while the diesel could reach 40%. The cost of gasoline was more than three times that of diesel fuel. Today diesel fuel is still only half the price of gasoline, but the efficiency figures have moved much closer together. This trend was noticeable in the early thirties, when aviation writers were inclined to dismiss the earlier attempts at practical aircraft diesels with a wave of the pen, opining that the high specific weight and complexity of the diesel fuel system effectively barred it from aircraft application. Junkers followed the MO-3 with an in-line five cylinder, liquid cooled, two cycle engine in 1935, and from then until World War II led the European air diesel field by periodically refining and enlarging this basic engine. The most powerful of the series was the Jumo 206, which developed 1,200 HP at 3,000 RPM—the fastest crankshaft speed ever attained in an aircraft diesel. This plant had a specific weight of 1.4 lbs./HP which was comparable to good gasoline engines of the day, and certainly an accomplishment for a water-cooled diesel. Junkers workhorse, the Jumo 205, flew as many air miles as many popular Amer-

ican gasoline engines from 1935 to 1943. This model was a six cylinder, in-line plant taking off at 700 HP with a specific weight of 1.72. In its nine year active life, it served as standard equipment on six types of multi-engined transport planes, four flying boats, two heavy bombers, and a dive bomber.

The diesel development program in Germany was divided between heavy transport craft and lighter-than-air ships. Daimler-Benz produced a six-cylinder, in-line engine for the largest of Germany's dirigibles. Designated the DB-602, the engine had a specific weight of 3.3 lbs./HP. Four of these bulky plants powered the Hindenburg, and four more drove the Graf Zeppelin.

While Germany was concentrating on in-line engines, Great Britain and the United States turned their interest to radial development. The American Packard diesels of 1928-31 seemed destined for a brilliant future, but were abandoned by the company after the death of the designer and chief proponent, Capt. Lionel Woolson. In May of 1931 Walter E. Lees and Frederick A. Brossy flew a diesel-powered Bellanca "Pacemaker" 84 hours and 32 minutes without refueling. The engine was a Packard, and the mark stands as an endurance record for both gasoline- and diesel-powered aircraft. Although they were working with a four cycle engine, Packard engineers did the first creditable job of air-cooling the aircraft diesel. The nine cylinder radial engine was reputedly free of cylinder head hot-spots which had plagued earlier air-cooled efforts. In fact, the Packard engine ran considerably cooler than many radial diesels developed after it. The engine was evidently highly regarded at the time of its designer's demise for the company received the Collier Trophy in 1931 for its development.

No sooner had the Packard engine left the scene than a smaller unit of similar design, the Guiberson A-980, appeared to take its place. This 185 HP, nine cylinder, radial diesel was the prototype for a series of Guiberson models producing up to 310 HP. This has been America's most versatile aircraft diesel, holding a Navy test contract for some time and appearing as a replacement engine in light Stinson and Waco private aircraft.

The Guiberson was the first medium-powered engine to operate successfully with air cooling, but its most important contribution was a dependable fuel injection system. Even the smooth-running Jumos had had some difficulty with this factor, but the Guiberson variable-stroke pump brought a flurry of refinements to other diesel fuel injection systems.

One of the most significant pre-war trials emerged from Great Britain. The Bristol Aeroplane Co. began work on an aircraft diesel, designated the Phoenix, in 1937. The design of this engine was borrowed from Bristol's contemporary gasoline model, the Jupiter VIII-F, and the sea level performance of the two models was almost identical. These engines were tested simultaneously in Westland observation aircraft; the diesel showed 50% less relative power loss at 20,000 ft., 15% greater rate of climb, and 35% reduction in fuel consumption. The Phoenix was an unsupercharged four cycle radial engine. It uncovered one important fact concerning aircraft diesels. The inherent low RPM of the engine could produce a maximum speed of only 110 knots at sea level in the Westland airplane, compared to 130 knots for the unsupercharged Jupiter. At 15,000 feet, however, the maximum

speed of the diesel craft was 120 knots while the gasoline engine had fallen off to 105. At that time, diesel superchargers were as efficient as gasoline blowers. In the turbo-superchargers of today, the diesel would hold a slight advantage because of the lower exhaust gas temperatures. The superiority of the diesel as a medium-altitude engine was proved in flight by these tests, and has since been corroborated in several instances. Despite the favorable indications, Bristol was reluctant to press the issue, and their diesel development program was of a secondary nature in the years before World War II.

England equipped two major dirigibles, the R-100 and the R-101, with diesel power in the early thirties. The engines were in-line, steam-cooled, four cycle power plants built by William Beardmore and Co. Although they performed very well in flight, their high specific weight (7.8 lb./HP) prevented their being adapted for use in lighter-than-air craft other than the heaviest dirigibles, or in any heavier-than-air craft. Napier and Sons contributed another British prewar entry, actually a tribute to the Junkers diesels. The Napier "Culverin" and "Cutlass" were licensed from Junkers in 1935, but never reached full production after several disappointing tests. Attempts to make direct diesel conversions of several Rolls-Royce "Condor" gasoline units ended on a rather dismal note when these engines developed only half the power of the original design.

France showed a particular interest in both two and four cycle radial diesels, producing eleven domestic models from 1930 to 1938, but none of these were ever thoroughly test flown. Despite their lack of practical accomplishment, the French did reduce the problems of cooling the radial two cycle engine. The Salmson SH-18, built under Czech license, was a combination air- and water-cooled eighteen cylinder affair, producing 600 HP at 1,700 RPM with a specific weight of 1.9 lb./HP. Incorporation of air-cooling was made possible by a unique auxiliary blower and loop scavenging system. This principle was furthered in a fully air-cooled two cycle engine built in 1937 by Botali et Cie. The fact that the engine never left the test stand was due largely to injection difficulties, and not to lack of proper cooling.

In the years preceding World War II, Germany stood alone in making diesel-powered flight economically feasible on a large scale. A long series of Junkers airliners employed the Jumo engines from 1928 through the war years. Germany's most successful heavy bombers were adapted from the JU-86 airliner and were diesel-powered. Blohm and Voss and Dornier flying boats, both commercial and military, used the Jumo 205. Junkers and Focke-Wulfe dive bombers spearheaded the early blitzkreigs on diesel-ized wings.

The advent of jet and rocket propulsion has understandably lulled aviation diesel development in the United States since the war, but the inherent advantages of compression ignition have been hotly pursued abroad. In West Germany, MAN has developed a basic engine capable of adaptation to truck, boat, or light aircraft use. The smaller model produces approximately 150 HP at only 1.5 lb./HP, consumes only 0.30 lb./HP-hour of straight aviation diesel oil, and will operate on almost any modern fuel, including 80-octane gasoline and aviation kerosene. The basic MAN type is an air-cooled

two cycle engine and promises to be the first wholly successful model of its kind. Cooling is effected by a turbo-cooler blower operated in tandem with the turbo-supercharger. The efficiency of the unit is remarkable; MAN seems to have no cooling problems in the engine. Engine temperature is also controlled by a radically new combustion progression in the cylinder. The combustion chamber is semi-spherical, and fuel injection is a two-stage process. The initial spray merely wets the walls of the chamber, followed by the ignition spray in the form of the conventional "cloud" of oil filling the chamber. The system produces a controlled combustion rate, but does not utilize an auxiliary injection system. Instead, the "wall-wetting" fuel is introduced as a cloud and thrown onto the cylinder walls by swirling air from the scavenging cycle. Near the end of the injection period, scavenging ceases and the resulting turbulence of gases in the chamber maintains the cloud of fuel. There is no evidence that larger models of this engine, adapted for aircraft, should not produce over 2,000 HP at reduced specific weight and specific fuel consumption. The advanced design of the scavenging system brings the two cycle diesel into direct competition in efficiency with gasoline engines.

The MAN engine is only a prototype. It was not originally designed for aircraft use and has not been thoroughly tested for this purpose. It will probably not find its greatest employment as an aircraft engine, but it does bring out two significant points. Of greatest importance, it now appears feasible to adapt air-cooling to the two cycle diesel and still retain thermal efficiency. Secondarily the barrier of slow and unreliable starting and long warm-up has been passed by use of an improved flame starter and an enlarged blower capacity. The developments of MAN in Germany are still open to speculation, but speculation of a very positive nature.

The German developments are supplemented by more tangible evidence of success in England. Napier, the firm which undertook the production of Junkers engines before the war, has produced a twelve cylinder, two cycle, liquid cooled diesel engine compounded with a gas turbine and designated the "Nomad." In the general field of compound engines, the diesel is basically more satisfactory than the gasoline engine for the following reasons:

1. Cylinder pressures in the diesel can be controlled by the rate of fuel input.
2. Exhaust back pressure becomes a help rather than a hindrance when high degrees of supercharging are employed.
3. In the absence of detonation and preignition troubles the compression ratio and degree of supercharging may be varied to suit design requirements.
4. The excess air over that used for combustion which passes through the engine may be used for cooling. The only limit to the amount of excess air which can be introduced is the critical size and weight of the compressor which moves the air.

The Nomad is unique in that it is the first aircraft diesel to be so compounded, but this is not the major point of interest. Although the Nomad at 1.03 lb./HP is 25% heavier than gasoline engines of the same power, the fuel consumption of this diesel is 40% lower. Exhaustive comparative tests for endurance and reliability have shown the Nomad to be the equal of the most

modern gasoline compounds in such former diesel downfalls as cylinder head hot-spots, fuel injection failure, and compressor difficulties from rapid changes in altitude. The engine's statistics are a refreshing improvement in aircraft diesel development. It produces 4,000 brake horsepower at 2,050 RPM, with a compression ratio of 27:1. The specific weight of the complete engine is 1.1 lb./HP. The gas turbine drives the blower and also augments engine power through an infinitely variable gear through a quill shaft to the crankshaft. At full power settings, the turbine produces 2,250 HP, of which 1,840 are absorbed by the compressor. Through the turbine coupling, the diesel may drive the compressor when maximum cruising economy is desired. The gas turbine will run on any fuel compatible to the diesel with the exception of gasoline over 100-octane, which produces exhaust gases too hot for proper turbine operation. Napier has successfully introduced water injection into the two cycle diesel, and this system gives the Nomad an additional 400 HP for emergency use.

It should be emphasized that the Napier Nomad, a liquid-cooled engine, is designed primarily for use on air freighters. It would not be readily adaptable for use on Navy patrol craft, although it shows definite possibilities in the field of military transport. Most significant is the added reliability shown by the Nomad at a definite reduction in specific weight.

One may begin to draw together the threads of diesel aircraft developments into a picture prophetic for naval aviation. The high-powered, air-cooled, two cycle diesel is no longer the subject of ridicule, and is much closer to realization than our more spectacular goals of Mars or Venus. The medium-powered two cycle plant is a reality; its natural advantages to observation and scout craft are undeniable. These diesels can reduce fuel consumption 30% or increase range the same amount. Endurance is at a peak, elaborate weather guards are unnecessary, engine operation is smoother, and climatic conditions have little effect on fuel or engine.

Patrol aircraft and low-altitude bombers could benefit greatly by the perfection of the high-powered two cycle diesel. Increased range and payload are the primary considerations, but also of importance are reduction of fire and explosion hazards, greater safe periods between engine check and overhaul, and continued operation of the engine after one or more cylinders have failed. In the transport of supplies and personnel aircraft diesels offer drastic reduction in fuel costs and improved safety on long flights, in addition to advantages previously mentioned.

One cannot assume that the aircraft diesel for naval use has fallen into our laps. This is definitely a long range development program. There are still major cooling problems in high-powered two cycle engines. The reliability of high capacity, multi-stage blowers leaves something to be desired. A good aircraft diesel, alone or compounded, offers too much to be lightly dismissed, however. There is clear evidence that the aircraft diesel can strengthen a great part of naval aviation, and thereby strengthen the whole. The challenge is fully as real as that of greater speed and is worthy of due consideration.

A NOMOGRAM FOR DETERMINING THE ENDURANCE OF NAVAL VESSELS

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received his commission from the United States Naval Reserve Midshipman School at Columbia University in November 1945.

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ABSTRACT

The purpose of this paper is to provide forces afloat with a method for readily determining their vessel's endurance depending on the amount of fuel oil remaining. The use of this method will provide accurate and rapid evaluation of varying the vessels speed. The result then will furnish forces afloat with a quick method of selecting appropriate vessel speed to reach their destination based on operational, economical, and time requirements.

INTRODUCTION

Modern Naval operations bring together vessels of many different fuel capacities and require wide flexibility in time underway. The need for quick and accurate knowledge of each vessel's endurance is well known to those concerned. Each and every vessel's remaining endurance is a variable in the successful completion of a Task Force Operation. Impending storms requiring the Task Force to get clear, failure of tankers to arrive, and expeditious departure from an area due to wartime exigencies are but a few of the situations where endurance becomes ever so important.

DEVELOPMENT OF NOMOGRAM

A parallel-line nomogram may be utilized to great advantage in solving this problem as the endurance of a vessel may be expressed as a mathematical equation. This equation is of five variables so that determinants and scale

factors were utilized to facilitate the construction of a suitable nomogram.

This development of the nomogram will be given so that operating personnel may construct a nomogram applicable to their vessel by following this procedure. It is necessary to have certain characteristic curves of the vessel on hand. These curves must show the relations between vessel's speed with shaft horsepower and fuel rate. A hypothetical set was taken for this development and is shown in Fig. 1.

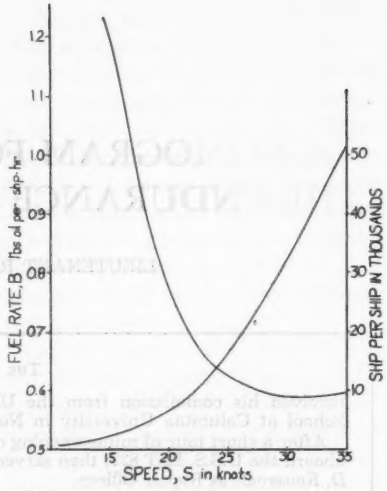


Figure 1

The equation

$$E = 8.33\frac{F\,S}{A\,B}$$

may be used to calculate the vessels endurance when:

E=endurance in nautical miles.

F=fuel oil on board, gallons.

S=speed in nautical miles per hour.

A=shaft horsepower.

B=fuel rate, lbs. per shp-hr.

8.33=a constant to provide dimensional homogeneity.

From the curves of Fig. 1, it was desirable to select the ranges of S, B, and A. Accordingly they are:

$$15 \leq S \leq 35$$
$$.59 \leq B \leq 1.2$$
$$2500 \leq A \leq 53000$$

The amount of fuel oil remaining can be any value up to storage capacity. In this solution the diesel and fuel oil capacities were combined to give an upper value of fuel on board of 155,000 gallons. The upper value of endurance may be calculated from the given relation using maximum fuel capacity, most economical speed, and corresponding fuel rate and shaft horsepower. In this solution this upper value of endurance was taken as 6450 miles, therefore:

$$1 \leq F \leq 155000$$
$$1 \leq E \leq 6450$$

The problem can be written in the form $f(F,t_1,S,B,t_2,E)=0$ where t_1 and t_2 are dummy variables. This is done by taking logarithms of both sides of the equation, setting equal to zero, and introducing the dummy variables by formulation of three new relations.

Thus $E=8.33\ F\ S/A\ B$ becomes:

$$\log F+\log S+\log 8.33-\log A-\log B-\log E=0$$

Then introducing the dummy variables gives:

(1) $\log F+(\log S+\log 8.33)-\log t_1=0$

(2) $\log t_1-\log B-\log t_2=0$

(3) $\log t_2-\log A-\log E=0$

Utilizing determinants and scale factors:

from (1)

$$\begin{vmatrix} -a_1 & b_1 \log F & 1 \\ 0 & \frac{b_1 b_3 \log t_1}{b_1 + b_3} & 1 \\ a_3 & b_3(\log S + \log 8.33) & 1 \end{vmatrix} = 0$$

from (2)

$$\begin{vmatrix} -a_4 & b_4 \log t_1 & 1 \\ 0 & \frac{b_4 b_6 \log B}{b_4 + b_6} & 1 \\ a_6 & -b_6 \log t_2 & 1 \end{vmatrix} = 0$$

from (3)

$$\begin{vmatrix} -a_7 & -b_7 \log A & 1 \\ 0 & \frac{-b_7 b_9 \log t_2}{b_7 + b_9} & 1 \\ a_9 & -b_9 \log E & 1 \end{vmatrix} = 0$$

The a scale factors give the desired width to the nomogram. The b scale factors give the desired height to each variable. Choosing the size of the desired nomogram permits solving for these scale factors. In this solution the size was chosen as 14″ x 18″.

Then from $b_1 \log F$
$14 = b_1(\log 155000 - \log 1)$;

$$b_1 = \frac{14}{5.19} = 3$$

Then from $b_3 \log S$
$14 = b_3(\log 35 - \log 15)$;

$$b_3 = \frac{14}{.368} = 38$$

Then from $b_7 \log A$
$14 = b_7(\log 53000 - \log 2500)$;

$$b_7 = \frac{14}{1.326} = 10$$

Then from $b_9 \log E$
$14 = b_9(\log 6450 - \log 1)$;

$$b_9 = \frac{14}{3.81} = 3.7$$

Then from $b_4 = \frac{b_1 b_3}{b_1 + b_3}$;

$$b_4 = \frac{(3)(38)}{41} = 2.78$$

Then from $b_6 = \frac{b_7 b_9}{b_7 + b_9}$;

$$b_6 = \frac{(10)(3.7)}{13.7} = 2.7$$

Then $$\frac{b_4 b_6}{b_4 + b_6} = \frac{(2.78)(2.7)}{5.48} = 1.37$$

A rough sketch will now show that a suitable shape is obtained by taking:

$a_1 + a_3 = 11$ and $a_7 + a_9 = 5$

and from

$a_1 b_3 = a_3 b_1$
$a_1(38) = a_3(3)$
$38a_1 - 3a_3 = 0$
$3a_1 + 3a_3 = 33$
$41a_1 = 33$
or $a_1 = .805$
and $a_3 = 10.195$

also

$a_7 b_9 = a_9 b_7$
$a_7(3.7) = a_9(10)$
$3.7a_7 - 10a_9 = 0$
$10a_7 + 10a_9 = 50$
$13.7a_7 = 50$
or $a_7 = 3.65$
and $a_9 = 1.35$

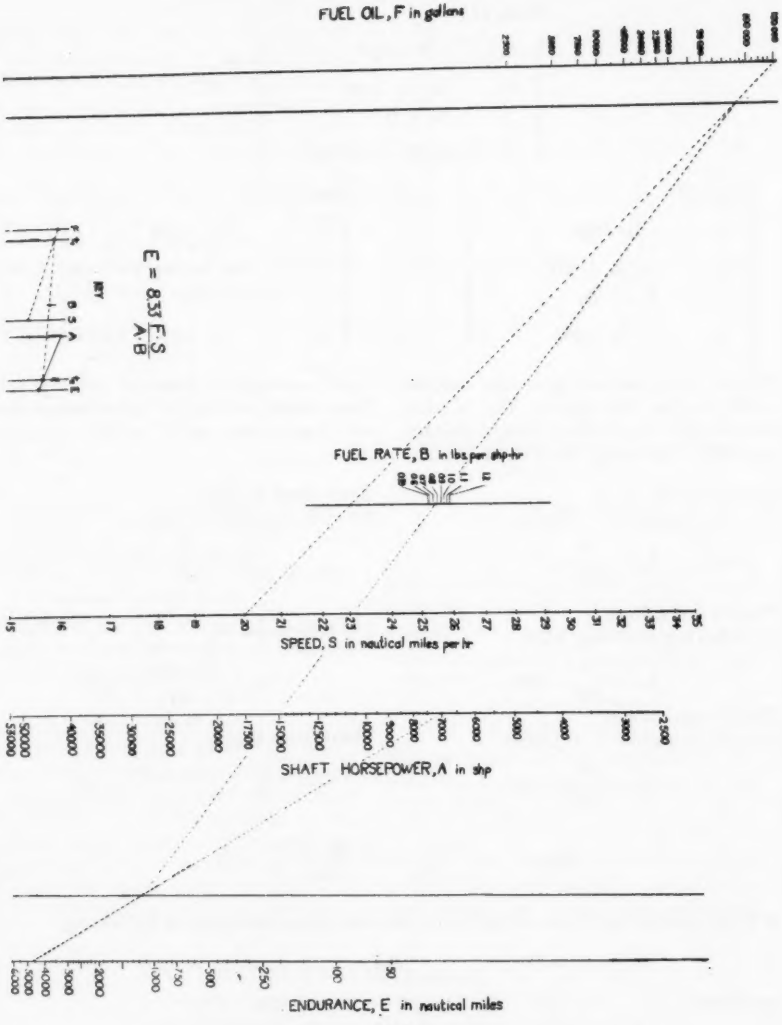

FUEL OIL, F in gallons
E = 8.33 F.S / A·B
FUEL RATE, B in lbs. per shp-hr
SPEED, S in nautical miles per hr
SHAFT HORSEPOWER, A in shp
ENDURANCE, E in nautical miles

Figure 2. The Nomogram

Now since the overall width is to be 18″

$$a_1 + a_4 + a_6 + a_9 = 18$$
$$.805 + a_4 + a_6 + 1.35 = 18$$
$$\text{or} \quad a_4 + a_6 = 15.845$$
$$\text{and also } a_4 b_6 = a_6 b_4$$
$$a_4(2.7) = a_6(2.78)$$

therefore $a_4 = 8.04$ and $a_6 = 7.805$

The determinants now become:

(1)

$$\begin{vmatrix} -.805 & 3\log F & 1 \\ 0 & 2.78\log t_1 & 1 \\ 10.195 & 38(\log S + \log 8.33) & 1 \end{vmatrix} = 0$$

(2)

$$\begin{vmatrix} -8.04 & 2.78\log t_1 & 1 \\ 0 & 1.37\log B & 1 \\ 7.805 & -2.7\log t_2 & 1 \end{vmatrix} = 0$$

(3)

$$\begin{vmatrix} -3.65 & -10\log A & 1 \\ 0 & -2.7\log t_2 & 1 \\ 1.35 & -3.7\log E & 1 \end{vmatrix} = 0$$

From which we have all the necessary information needed to construct the nomogram. The construction is started by drawing a vertical line near the left hand edge of the paper. Along this line the scale of 3logF (see Appendix) is laid down from the lower limit at the bottom to the upper limit at the top. At a distance of .805″ to the right another vertical line is drawn, this is the t_1 line or index line and requires no scale. Then at a distance of 10.195″ to the right of this index line another vertical line is drawn; along this line lay off the scale 38(logS −log8.33), starting with the lowest graduation at the intersection of a horizontal perpendicular line from the lower limit of the F scale. At a distance of $(a_7 + a_9)$ 15.845″ to the right of the first index line place another vertical line to become the t_2 line or second index line. Then at a distance of 3.65″ to the left of the second index line place another vertical line and lay off the A scale (−10logA) with the greatest value down and at the intersection with the previous perpendicular as it is negative in the determinant. Then at a distance of 1.35″ to the right of the second index line place another vertical line and lay off the E scale (−3.7logE) with the greatest value down as it too is negative in the determinant.

It then remains to work out the B scale. At the point of intersection of the first index line and the perpendicular draw a line at any convenient angle (say 45°) with the horizontal perpendicular and lay off the distance $(b_4 + b_6)$ 5.48″. Join this end point with the point of intersection of the second index line and horizontal perpendicular. Then drawing a line parallel to this last line pass it through a point laid off a distance (b_4) 2.78″ up the line drawn at 45°. At the point where this parallel line intersects the horizontal perpendicular erect another vertical line for the B scale. The maximum and minimum values of B can then be located by solving the relation E=8.33 F S/A B; for each value of B use corresponding values of S and A, total fuel capacity, and solve for E. Then connecting points on scales previously drawn the line between index lines intersects the B line at the appropriate value of B. Between these maximum and minimum

locations lay off the B scale (1.37logB). The nomogram is now ready for checking.

CONCLUSION

The procedure of this paper requires curves which are available or may be prepared by forces afloat. A nomogram of this type will give the ships endurance only after the remaining fuel oil is known. This in itself is time consuming. However, once known, the nomogram offers a rapid method of evaluating the endurance for various speeds without calculations.

It is entirely possible that further work along this line could produce other nomograms which would be useful throughout the Naval service. Future nomograms might include:

(a) In lieu of providing the Task Force Commander with a nomogram from each vessel develop a multiple scale nomogram adaptible to vessel types by choice of scales.

(b) Modify the nomogram of this paper for more readable endurance values in the lower range so that with steady steaming in joining up with a tanker, the tanker can easily be previously informed how much fuel will be needed.

(c) Make allowances to shaft horsepower due to bottom fouling.

For larger vessels it appears that pairing off fuel oil remaining with shaft horsepower would give a more suitable nomogram instead of fuel oil remaining with speed as in this paper.

ACKNOWLEDGEMENT

The author is indebted to Captain R. E. Bourke, USN for the idea of the Endurance Nomogram.

REFERENCES

"The Nomogram" by H. J. Allcock, T. Reginald Jones and revised by J.G.L. Michel, Pitman Publishing Company.

APPENDIX

Procedure for laying off scales.

Select values of F desired to have marked off and make up a table.

Having the vertical line drawn call the distance up this line y measured from the horizontal perpendicular.

F	logF	y = 3log F
1	0	0
2500	3.400	10.2
5000	3.699	11.1
7500	3.875	11.63
....		
....		
....		
....		
150000	5.176	15.54
155000	5.190	15.58

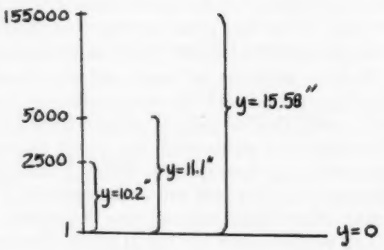


Figure 3

ACOUSTICS AND NAVAL ENGINEERING

ROBERT TAGGART

THE AUTHOR

recently was in charge of the Noise Section, Applied Science Division, of the Bureau of Ships. He received his Bachelor of Science degree in Naval Architecture and Marine Engineering from Webb Institute of Naval Architecture in 1942. He served with the Consolidated Steel Corporation, U. S. Maritime Commission, and U. S. Army Transportation Corps prior to his employment in the Navy Department. His Navy Department position included the administration of research investigations in the fields of airborne and underwater noise reduction of naval vessels. At the present time he is Chief, Analytical Engineering Department, Reed Research, Inc.

INTRODUCTION

Although at first glance the fields of acoustics and naval engineering appear only remotely related, that relationship is continually becoming more intimate. In all naval vessels, habitability considerations demand low levels of noise and vibration. On aircraft carriers, the excessive noise of jet planes causes aural injury to flight deck personnel and interferes with bridge communications. In engine rooms, the trend toward high speed, light weight machinery is tending to increase noise to the deafness level. Underwater noise interferes with the operation of underwater communication equipment and in anti-submarine vessels it reduces sonar effectiveness. Radiated underwater noise reveals the presence of the otherwise invisible submarine and renders all vessels susceptible to the insidious acoustic mine and acoustic torpedo.

Thus it is important that the naval engineer understand the fundamentals of acoustics in order that the basic concepts of this branch of engineering may be taken into consideration in the design, construction, and operation of naval vessels. This article attempts to express some of these fundamental principles and to show their application to one of the major naval problems —the detection and detectability of submarines.

NOISE AND UNDERSEA WARFARE

The ability to listen for other ships and to remain undetected while accomplishing a mission is of greater importance to the submarine than to any other type of vessel. Good listening capability permits detection of enemy

vessels at greater ranges, which places the submarine in a better position to evade or to launch an attack. Quietness is the most effective countermeasure against enemy detection equipment. The military effectiveness of a submarine is entirely dependent upon its capacity to remain hidden from the enemy. Radar, the airplane, and other means of surface detection have forced the submarine to remain totally submerged in order to realize its military potential. Thus the submarine must rely on its "ears" to fight and must present a minimum signal to the "ears" of the enemy.

These two facets of submarine noise, that of detecting and that of remaining undetected, are generally separated into two distinct categories. Noise which interferes with a ship's own sonar is categorized as "self noise." Noise which subjects a vessel to detection by another vessel is categorized as "radiated noise."

Self noise can consist of hydrodynamic noise, machinery noise, hull noises and propeller noise. Hydrodynamic noise is generated by flow around the hull and appendages, through the superstructure, and past the sonar transducer. Machinery noise originates in any machine with moving parts and passes through the hull structure to the water and thus to the sonar transducer, or directly to the sonar transducer as a structural vibration. Hull noises can be caused by the working of the hull structure, the sympathetic vibration of loose or flimsy equipment, or by the rattling of loose gear in the superstructure when excited by flow forces. Propeller noise can be generated at low speed by the pressure variations as the blades pass close to the hull, by thrust and torque variations on the shafting system, or by vibration of the propeller blades. At cavitation speeds a loud crackling noise is generated by the collapse of air cavities.

The most important components of radiated noise are generally cavitating or singing propellers, main propulsion machinery noise, and to a lesser extent, auxiliary machinery noise. Hull noises and hydrodynamic noise are not generally a problem from the radiated noise standpoint.

When an anti-submarine surface vessel is searching for or attacking a submarine, echo ranging, or active sonar is employed. In this type of sonar a repeated, high frequency acoustic signal is sent out from the sonar transducer. If this signal strikes an underwater object, it will echo back and be received by the transducer. The time difference between the sending and the receiving of the signal is a measure of the target range, and the direction from which the echo comes is a measure of the target bearing.

For long range detection of a target, active sonar is not satisfactory. The loss of signal strength, or attenuation, in sea water is excessive for high frequency directional signals. In submarine versus submarine warfare the use of active sonar gives away the presence of the attacking vessel to the target. Thus for long range detection, listening (or passive) sonar is employed. Passive sonar consists of a transducer, or underwater microphone, an amplifying system, and a speaker or earphones. Through this system the underwater noise radiated by the target can be heard. Generally two or more transducers separated by a fixed distance are employed. These transducers are mounted in a horizontal plane and by measuring the phase difference in the signals received, the direction of the signal can be computed. In some types of passive sonar the transducer assembly can be rotated mechanically until the phase difference is zero, thus training the unit in the direction of the target. In other types the training is accomplished electrically with the transducers remaining fixed.

Since transducers of this type are generally omni-directional, the zero phase difference position could indicate a target in either of two positions 180° apart. To prevent such confusion the back face of the transducers is heavily baffled so that the noise arriving in the forward face of the assembly will far exceed that arriving at the after face. In other words, the baffling increases the front-to-back discrimination.

The mechanically trained passive sonar system is very similar to human ears. The ears are separated by a fixed distance and can be trained in the direction of an incoming sound by turning the head until the phase difference in the arriving signal is zero. The ear horn serves as a baffling system to increase the front-to-back discrimination.

The human ear is also important in the overall performance of a passive sonar system. This performance is dependent upon the ability of a sonar operator to distinguish the sound of a target from the noises produced by his own ship. His ability to recognize certain sounds helps to identify the target. By discerning differences in frequency he can count propeller beats and often guess the speed of a target, and his estimate of relative loudness of a sound can help determine the bearing and range of a target. Thus the physics, physiology, and psychology of hearing are of extreme importance to submarine and anti-submarine warfare.

HEARING

The human ear is divided into three general regions as shown in Fig. 1—the outer, middle, and inner ears.

The outer ear forms a horn, open to the air. The outer lip of this horn, the *pinna*, serves to channel sound into the *external canal* and thence into the *eardrum*.

The middle ear is filled with air which is opened to atmospheric pressure through the *Eustachian tube* which connects it to the throat. This explains why a variation in atmospheric pressure such as that encountered in ascending or descending in an airplane can be balanced out by opening the mouth and swallowing.

The inner ear is filled with liquid and contains the nerve endings, the excitation of which causes the sensation of hearing.

The structure of the ear is such that airborne noise impinging on the eardrum is efficiently conducted to the liquid of the inner ear. This is accomplished through a mechanical linkage of small bones called the *ossicles*. This linkage connects the eardrum to the membrane separating the middle and inner ear which is called the *oval window*. The *ossicles* are also a partial safety device which become disjointed when the eardrum is subjected to extremely high amplitudes. Essentially they perform the function of an impedence matching transformer between the air and liquid-borne sound.

The inner ear consists of the *cochlea* and the *semicircular canals*. The latter are not described here since they are a

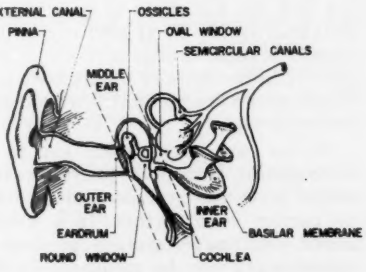


Figure 1
Structure of the Ear

part of the stabilizing mechanism of the body and have little connection with hearing.

The *cochlea* is a spiral tube a little over an inch in length. It is divided into two galleries by the *basilar membrane* which is a strip of nerve endings. One of the galleries is terminated at the inner-middle ear boundary by the *oval window* and the other by the *round window*. The vibration of the *oval window* by the *ossicles* forces the fluid in one gallery into vibration which is transmitted through the *basilar membrane* to the other gallery and thence to the *round window*. The *basilar membrane* with its connecting nerves is the microphone, frequency analyzer, and sound level meter which receives and interprets sound pressure fluctuation and transmits information to the brain.

We know a good deal about the behavior of the ear as a result of many subjective experiments. This knowledge is generally sufficient for the purpose of understanding how the ear will react to underwater sounds heard through a sonar system.

The young normal human ear can hear sounds at frequencies between 20 and 20,000 cycles per second. It can detect, as perceptible change in the pitch of a pure tone, a change in frequency of less than ½ percent. The ear is most sensitive at frequencies between 1000 and 5000 cycles per second; the range between the threshold of hearing and the threshold of pain is approximately 120 db at 1000 cycles per second. A rapid change of less than one decibel in the level of a pure tone can be perceived at a comfortably loud listening level.

Earlier we likened the ear to a microphone, frequency analyzer, and sound level meter. This analogy may be carried even further. In a simple sound level meter system, the lowest signal which can be measured is determined by the microphonics and electrical self noise of the system; that is, the signal must be of an intensity greater than the self noise in order to be measured. This lower limit is termed the "threshold" of the system. The human ear also has a threshold which is determined by the self noise generated by muscular action and the movement of the circulatory and respiratory systems.

The microphone—sound level meter system also has an upper limit of usefulness. At high sound pressure levels the amplifiers become blocked and the meter will no longer function accurately. Permanent exposure to such levels can destroy the sensitivity of the instrument. The human ear also has an upper limit of usefulness. The level at which this occurs is called the "threshold of pain." If the ear is often exposed to levels of this magnitude, permanent deafness or loss of hearing can result.

Both the threshold of hearing and the threshold of pain vary with frequency as does the most comfortable listening level. These three levels are shown plotted against frequency in Fig. 2.

As noted before, the ear functions as a frequency analyzer. It can easily distinguish tones of different frequencies in the mid-frequency range and can generally give a fair measure of the relative quantitative value of two or more frequencies. For example, the average person can detect whether or not a piano is in tune by playing a series of octaves or basic chords. Some persons who have so-called "absolute pitch" can give absolute values of frequencies in the mid-frequency range when they hear a *single pure tone or a chord.*

These discriminatory powers decrease at the lower and higher frequencies. When playing the scale on the piano, the difference in pitch of the notes in the middle of the keyboard can be readily detected. However, at the upper and lower ends of the keyboard, these differences are harder to detect.

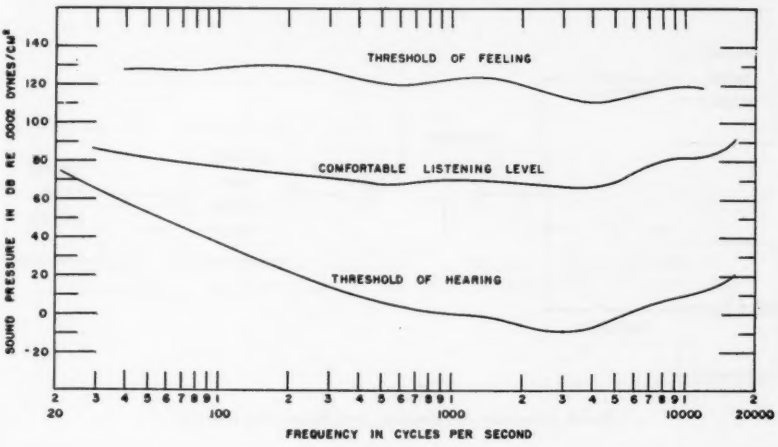


Figure 2
Response Curves of the Human Ear

RECOGNITION OF UNDERWATER SOUNDS

Unlike listening to a series of pleasant chords played on a piano at a comfortable listening level in a quiet room, the sonar operator is trying to hear and identify a complex signal through a group of masking noises. These noises consist of own ship's noise, system noise, and sea background noise. This is somewhat like trying to listen to a gravel-voiced announcer describing a baseball game over the noise of cheering rooters through a static-filled radio in a crowded barroom. Thus, the manner in which the ear functions in detecting a wanted signal in the presence of masking noises is of utmost importance.

As in the case of the baseball fan cited above, the ear can hear many sounds at once but can concentrate on the wanted sounds disregarding, to some extent, the masking background. In this respect, the ear functions somewhat as a band pass filter—accepting those signals within the desired band, and rejecting those outside of that band. This can result in better discrimination of a wanted signal in the presence of a masking background as shown by the following example:

Given a microphone, a sound level meter, and an octave band filter set as a receiving and analyzing system, and an oscillator, thermal white noise source and speaker as a sending system. A block diagram of these systems is shown in Fig. 3.

Now if switch A is closed and the filters set in the "overall" position, the broad band spectrum received by the sound level meter would look like that shown in Fig. 4A, assuming distinct cutoffs at 20 cps and 20,000 cps.

The level shown on the sound level meter would be equivalent to the total energy of this sound which is the same as the area under the curve shown (i.e., average power per cycle times the number of cycles in the audio frequency range). The 80 db can be converted to a linear scale and multiplied by 19980

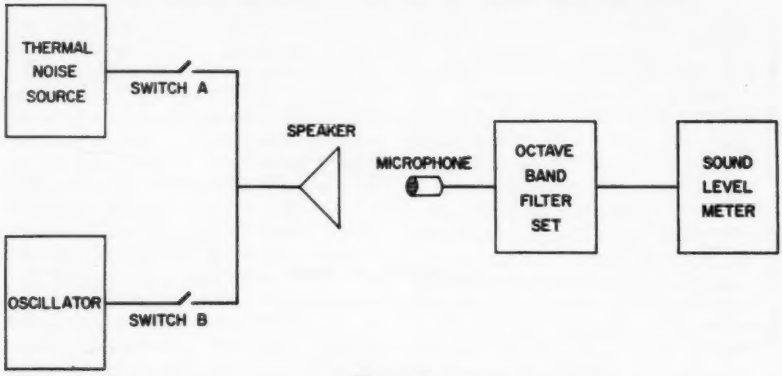


Figure 3
Block Diagram of Sending and Receiving System

cycles, but it is simpler to use the logarithmic multiplication 80+10 log (20000−20). This is equal to 123 db.

If switch A is opened and switch B closed and a 150 cps tone at 110 db is sent out from the oscillator the "overall" level read on the sound level meter would be 110 db and the spectrum would be as shown in Fig. 4B.

If both switches A and B are closed the combined frequency spectrum will be as shown in Fig. 4C. Considering the sound level meter's inability to readily detect small changes in sound power, it would not "hear" the presence of the pure tone and would still register 123 db.

Now if the octave band filter is set to pass the two octaves from 100 to 400 cps and the procedure repeated, assuming again distinct cutoffs imposed by the filters, the spectrum of the thermal noise source alone would appear as in Fig. 4D and the level registered on the sound level meter would be equivalent to the total energy of this noise of 80+10 log (400-100) =104.8 db.

If we then closed switches A and B, the combined frequency spectrum would be as shown in Fig. 4E. The pure tone would now dominate, and the combined level would be 111.2 db. Thus by the use of the filters, we have permitted the sound level meter to discriminate the signal from the masking background by a difference of approximately 6 db when the signal was introduced. If the experiment is continued and the filter pass band narrowed to between 100 and 200 cycles, another 4 db of discrimination is gained. Thus it can be seen that by narrowing the pass band through which a masking background is permitted to reach the sound level meter, increased discrimination of the desired signal is obtained.

The human ear works in a similar manner. The band of frequencies which it passes is termed the "aural critical band." This band, however, is not the same at all frequencies. It is much narrower at the mid frequencies of the audio frequency range than at the high and low frequencies. This bandwidth plotted against frequency is the result of many subjective tests and is shown in Fig. 5.

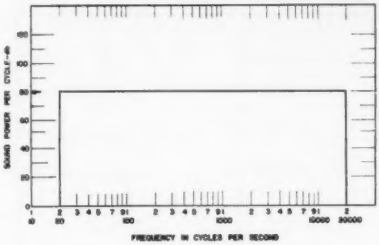


Figure 4. Sound Spectra
A, Thermal Noise at 80 db Level

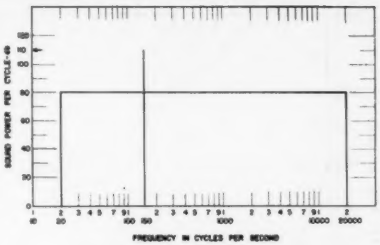


Figure 4. Sound Spectra
C. Thermal Noise and Pure Tone

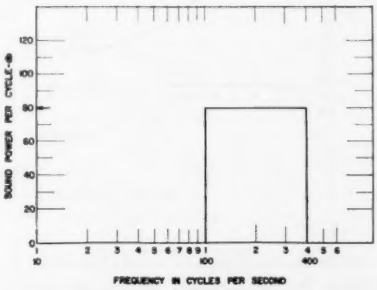


Figure 4. Sound Spectra
D. Thermal Noise through Band Pass Filter.

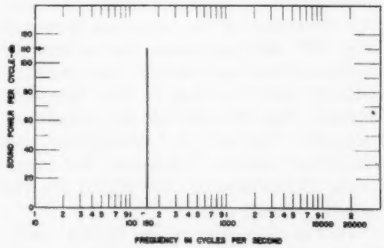


Figure 4. Sound Spectra
B. Pure Tone at 110 db Level

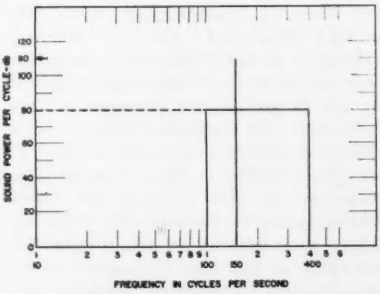


Figure 4. Sound Spectra
E. Filtered Thermal Noise and Pure Tone

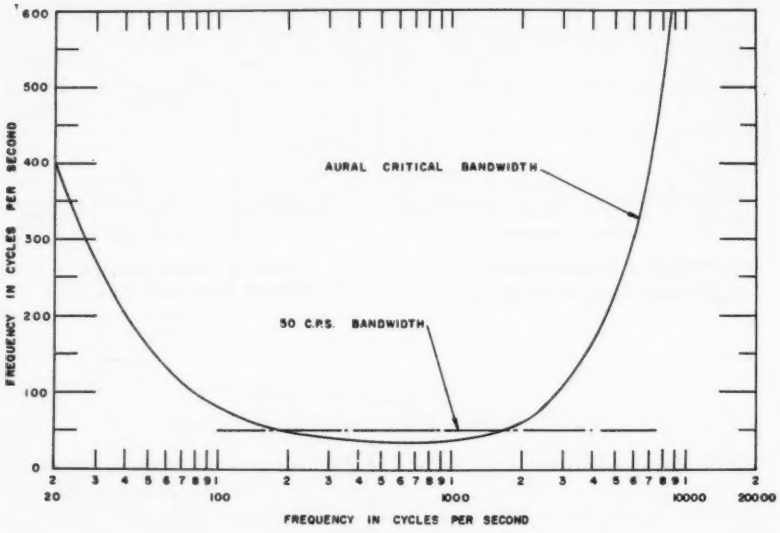


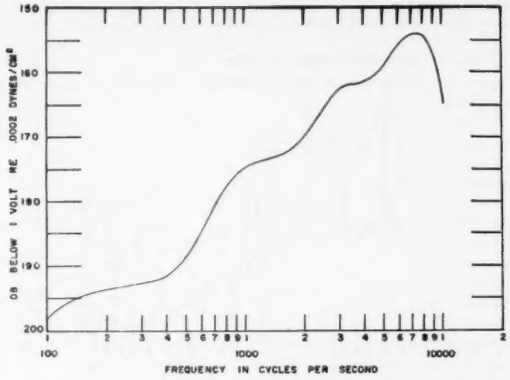
Figure 5
Aural Critical Bands of the Human Ear

In order to compare available instruments to the human ear, a constant value of 50 cycles per second may be used as a substitute for the aural critical band. It can be seen in Fig. 5 that this represents a fair average of those bands which are of general interest.

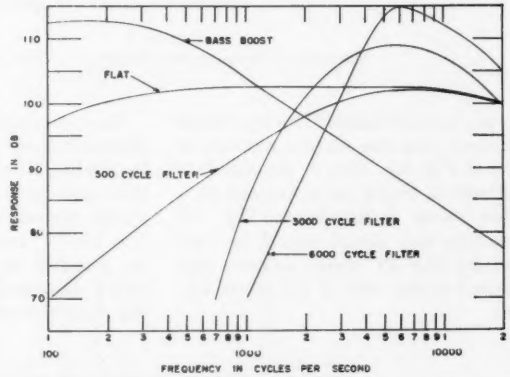
To return to the sonar operator, the underwater sounds heard by him are not directly comparable with the sounds produced. He is listening through a series of electronic components which distort the sound radiated by the target. As an example of this distortion the response curves of a standard passive sonar system, the JT, are given in Fig. 6. Fig. 6A shows the response of the hydrophone itself which tends to reshape the arriving signal in a manner which lowers the intensity of the low frequency components and emphasizes the high frequency components up to 8000 cycles per second. The amplifier circuit contains several elements which can vary the character of the signal as shown in Fig. 6B. The operator can select the response position which is most satisfactory for listening to the incoming signal. Fig. 6C shows the combined response curves of hydrophone and amplifier which represent the complete distortion of the signal by the sonar system.

In Fig. 7 is shown the typical spectrum of the underwater noise produced by a machine installed aboard a submarine. The spectrum is reproduced as it has been analyzed through two different sets of filters—one which hears the noise in bands which are one cycle per second wide, and the other which hears the noise through 50 cycles per second bandwidths. This latter value represents the aural critical band. Superimposed on the machinery noise spectrum is an average value of the sea noise spectrum in both one and 50

JT Hyrdophone
Response Curve



JP Amplifier
Response Curves



JT System
Response Curves

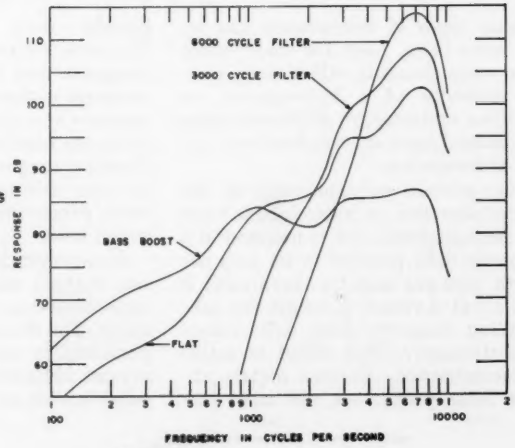


Figure 6
JT Sonar System Response Curves

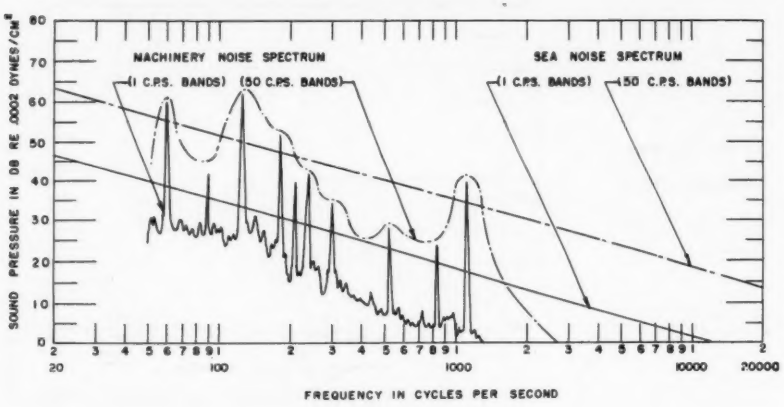

MACHINERY NOISE SPECTRUM
(1 C.P.S. BANDS) (50 C.P.S. BANDS)
SEA NOISE SPECTRUM
(1 C.P.S. BANDS) (50 C.P.S. BANDS)
SOUND PRESSURE IN DB RE .0002 DYNES/CM²
FREQUENCY IN CYCLES PER SECOND

Figure 7
Machinery Noise in the Presence of Sea Noise

cycle per second bands. The combined machinery and sea noise spectrum is shown in Fig. 8A. This is the absolute signal which might be presented to a passive sonar hydrophone. Fig. 8B shows how this signal would be distorted by the JT sonar system and presented to the ears of the sonar operator.

The sonar operator does not have the discriminatory powers of the one cycle bandwidth filter and therefore he loses the noise peaks between 250 and 900 cycles per second as shown in Fig. 8C. The part of the noise spectrum which he actually has available to him to make detection and identification are the shaded areas of the curve.

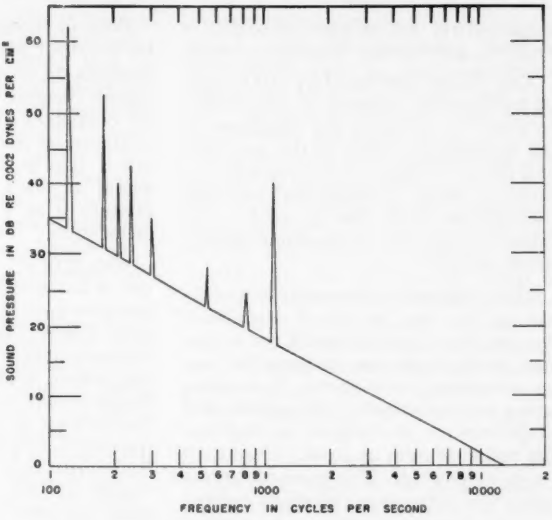
MEASUREMENT OF UNDERWATER SOUND

Many types of instruments can be, and have been, used for underwater noise measurements. All of them consist basically of a hydrophone, an amplifier, and a meter. Additional units such as analyzers or recorders may be a part of the system.

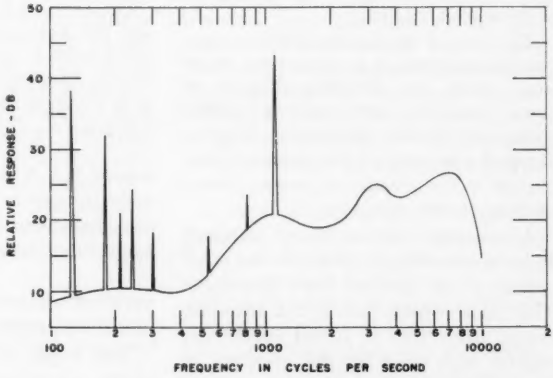
Hydrophones are generally of the magnetostrictive or piezoelectric type. If a ferromagnetic rod is inserted in a magnetic field parallel to its axis the length changes slightly. Inversely, if such a rod is varied in length the surrounding magnetic field will change proportionately. This effect is called *magnetostriction*. Various metals exhibit these properties, the most frequently used of which is nickel. Magnetostrictive hydrophones are so designed that the variation of sound pressure in the surrounding water will contract and elongate a series of nickel elements inserted in coils of wire. The changing magnetization of the elements induces alternating currents in the coils proportional to the impressed sound field.

Some crystals develop surface electric charges when subjected to pressure or tension. This is the *piezoelectric* effect. The developed charges are proportional to the force applied to the crystal. Materials used for hydrophone construction are Rochelle salt, quartz,

Machinery Noise
+ Se Noise
(1 C.P.S. Bands)



Machinery Noise
+ Sea Noise
+ JT Flat Response
(1 C.P.S. Bands)



Machinery Noise
+ Sea Noise
+ JT Flat Response
+ Ear Response
(50 C.P.S. Bands)

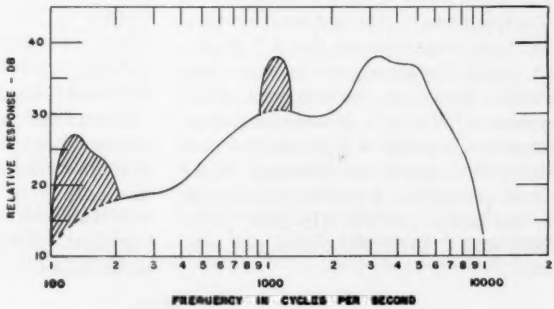


Figure 8
Machinery Noise and Sea Noise As Heard Through a Sonar System

ammonium dihydrogen phosphate (ADP), and barium titanate crystals.

The voltage produced by either type of hydrophone element is very small. In order to overcome the resistance of the hydrophone cable to the amplifier and meter, it is generally necessary to build a small preamplifier into the hydrophone itself. This is usually contained within the hydrophone waterproof casing.

The hydrophone-preamplifier output is fed into a set of amplifiers. Earlier, the output of the JT sonar system hydrophone and its amplifier response curves were shown. In a measuring instrument the hydrophone and amplifiers are so designed as to give a flat response over as great a frequency range as possible. This gives minimum distortion of the received signal and requires a lesser correction to be applied to measured results.

The output of the amplifiers is corrected as necessary to give meter readings which are absolute relative to some standard base such as 0.0002 dynes per square centimeter. Various stepped attenuators are generally included in the circuit to prevent overloading of the system.

A recently retired Navy standard sound measuring equipment—the OAY—was of the general type described. This unit, which had a flat response range from 100 to 10,000 cycles per second, was used for the majority of the underwater sound measurements of submarines for the last several years. Its recent replacement, the AN/PQM-1A sound measuring set, is somewhat similar. However, the response of the system is flat over a greater frequency range and a series of high and low pass filters have been incorporated. These filters permit band pass measurements in the bands 100-316, 316-1000, 1000-3160, and 3160-10,000 cycles per second.

Both of these sound measuring sets have output jacks to permit additional analysis or recording of the amplified signal. Several types of analyzers can be connected and the signal can be analyzed as the measurement is being taken. It is also possible to record the signal on discs or magnetic tape and analyze it at a more convenient time or location.

One type of analyzer currently in use is the *octave band filter set*, which contains a series of fixed high and low pass filters which permit band pass measurements in bands which are one octave wide. An octave, in sound measurement terminology, is a band of frequencies of which the upper frequency is twice the lower frequency. Standard filter sets of this type have cutoffs at 37.5, 75, 150, 300, 600, 1200, 2400, and 4800 cycles per second.

Other fixed filter types of analyzers are the half-octave analyzer and the third-octave analyzer. In the half-octave filter the upper cutoff of a band is $\sqrt{2}$ times the lower cutoff. In the third-octave filter the upper cutoff of a band is $\sqrt[3]{2}$ times the lower cutoff. A type similar to the third-octave filter set which is in limited Navy use is the Logit filter. In this filter set the upper cutoff of a band is $\sqrt[10]{10}$ times the lower cutoff frequency.

The types of analyzers described above are known as *contiguous-band analyzers*. The upper and lower cutoff filters are generally introduced into the meter circuit by push buttons or detented rotary switches.

Contrasted to these are continuous sweep types of analyzers one of which is the *heterodyne type*, which utilizes the output of a variable frequency oscillator which is added to the incoming signal and passed through a fixed band-pass filter. Nominal filter widths

of 2, 5, 20, 50, and 200 cycles per second are available. The oscillator frequency can be continuously varied and the output through the selected band width can be read at any desired frequency. In most of these analyzers a motor drive is attached to the oscillator frequency control so as to continuously analyze throughout the entire frequency range of the instrument. The motor drive is linked to a graphic level recorder to provide a complete frequency analysis.

A similar automatic analyzer is the *constant-percentage bandwidth* analyzer. Here the bandwidth is in constant proportion to the frequency being analyzed. For example, in a 10% bandwidth analyzer the bandwidth at 100 cps would be 10 cps and at 1000 cps it would be 100 cps. This analyzer can be manually or automatically operated.

Each of these types of analyzers has a useful place in the underwater acoustics field. In general it can be said that the type of analyzer to be used is a function of the degree of detail desired in a particular analysis, the initial and maintenance costs of the analyzer, the time required for analysis, the portability of the instrument, and the availability of a given analyzer for particular jobs. Personal preference of experimenters is also a factor which cannot be overlooked.

A comparison of the bandwidths available in many of these analyzers with the aural critical bands is given in Fig. 9.

It is highly desirable to know how measurements made with instruments compare with what a sonar operator can actually detect when listening to the same noises. The factor by which the instrument measurement is compared to the signal heard by the sonar operator is termed the *Recognition Differential.*

Assuming that sonar designers are successful in reducing system noise to a minimum and that the noise reduction program succeeds in lowering own ship's noise, the remaining masking noise which must be contended with is the ambient noise of the sea itself. This is a randomly fluctuating noise which varies with the state of the sea and which generally decreases in level with increasing frequency. In order to deal with it practically, many assumptions must be made. First, it must be assumed that only the ambient levels existing in the open ocean down to depths of a few hundred feet are to be considered. The average levels obtained from numerous measurements must be used rather than any specific measurement. Such averages have been obtained and are shown in Fig. 10.

It can be seen that these curves have a consistent slope of -5 db per octave and that the level varies with sea state. In dealing with the Recognition Differential, only the slope of the spectrum is of interest and not its level. Therefore, it can be assumed that the masking noise is continuous between 100 and 10,000 cycles per second (the range over which these measurements were made) having a slope of -5 db per octave.

A true Recognition Differential is obtained by subjective tests. Perhaps the most effective method of defining the Recognition Differential is to describe how a test is conducted. The detection of a signal in the presence of a masking background will not always occur at the same signal-to-noise ratio for different listeners or even for the same listener at different times. Therefore, several persons are employed for each test and many repeat runs are made. Assuming a test set up similar to that shown in Fig. 11, the noise source puts out a continuous noise of the same spectrum as sea noise at a comfortable listening level, say 70 db between 100 and 10,000 cycles per second. The signal generator is emitting a noise which may be the recording

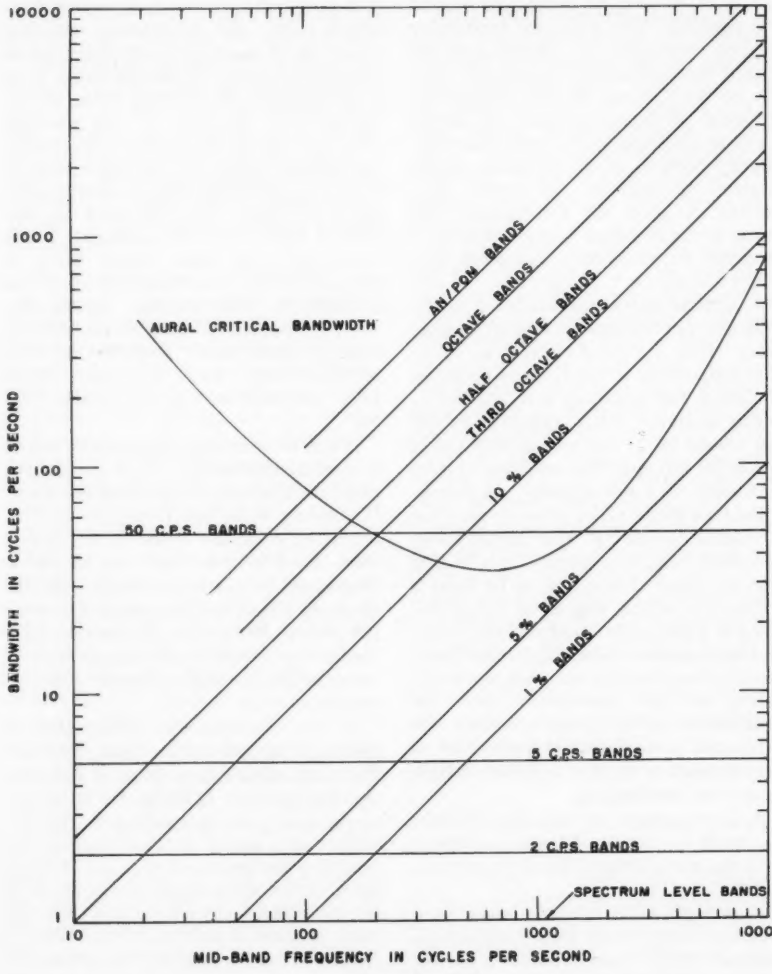


Figure 9
Bandwidths Available in Various Types of Frequency Analyzers

of a submarine trim pump. This signal can be varied in level by adjusting the attenuator. The actual level being played through the speaker can be read on the meter.

As many as ten observers may be used for the test. They will listen to the combined signal and noise being played through the speaker. The operator will then vary the signal at-

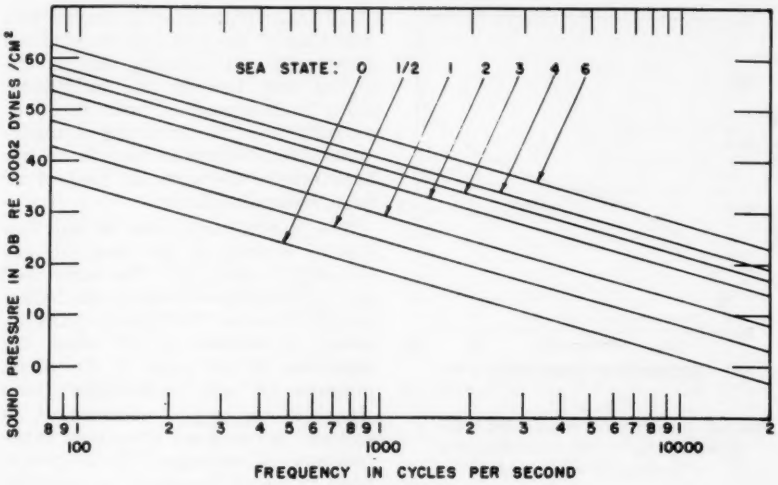


Figure 10
Average Spectrum Levels of Ambient Sea Noise

tenuator setting to different levels without indicating to the observers what that level is. At each setting he will call for a vote as to which observers can distinguish the signal from the noise. Each level is repeated several times. Assuming ten observers and five repeat runs for each level, there will be a total of 50 votes per level. The results of such a test are tabulated below:

Signal Level - db	Vote: Heard	Vote: Not Heard	Recognition Probability - %
60	0	50	0
61	0	50	0
62	2	48	4
63	9	41	18
64	16	34	32
65	32	18	64
66	43	7	86
67	49	1	98
68	50	0	100
69	50	0	100
70	50	0	100

These results are plotted in Fig. 12. Since the relative level at which detection occurs is variable, the Recognition Differential is defined on the basis of the relative level at which detection occurs 50% of the time. In the case of

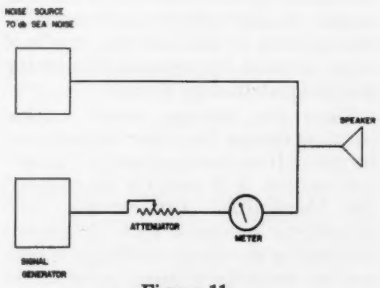


Figure 11
Test Set Up for Determination of Recognition Differential.

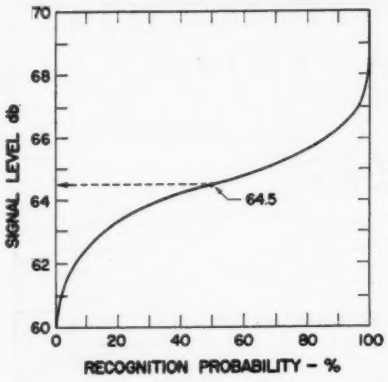


Figure 12
Plot of Recognition Probability for a Given Signal in the Presence of a Masking Background.

the test results plotted in Fig. 12, this occurred at a signal level of 64.5 db. The Recognition Differential is then the difference between the wide band level of the masking noise and the average level at which detection occurred, or RD=64.5−70=-5.5 db.

If this experiment were performed using a masking noise level of 50 or 90 db, the resulting Recognition Differential would have been the same. In other words, the Recognition Differential is independent of the absolute levels of signal or noise but depends only on the spectral distribution of each.

Since the average aural critical bands of the ear have been reliably determined from many subjective tests of this nature, it is possible to compute the Recognition Differentials with reasonable accuracy from frequency analyses of the noise recordings of submarine machinery using an assumed sea noise background.

Underwater noise levels, regardless of frequency, are generally reported in decibels relative to a sound pressure level of 0.0002 dynes per square centimeter. However, since the effectiveness of all sonar equipment is limited to a threshold of the spectrum level of sea state zero, this sloping line is the logical base line for all underwater noise levels associated with deep water conditions. The lowering of underwater noise of naval vessels to this base line is the ultimate goal of all noise reduction efforts.

The level of any noise at any frequency relative to this base line can be readily calculated. The term used for this calculated value is the *Detection Differential*. The Detection Differential is defined as 20 times the logarithm of the ratio of the sound pressure of any underwater noise signal to the sound pressure of an ambient noise of sea state zero at the frequency of the signal. The frequency band in which the signal is measured and the relative location of the point of measurement must be specified.

Since the primary concern at the present time is with interference in sonars utilizing aural presentation, the logical frequency band to select for this calculation is a fifty cycles per second bandwidth. When the noise signal is analyzed using filters of this bandwidth, the Detection Differential can be determined by subtracting the sea state zero level computed in the same bands.

Analysis in these narrow, constant bands is not always practical, however. For most work it is preferable to use wider bands such as third-octave bands, half-octave bands, or octave bands. These wider bands will not give as accurate knowledge of detectability, but will frequently be easier to handle where the analysis of many measurements is involved.

Although it is possible to convert levels measured in wider bands to equivalent levels in narrow bands, the procedure is subject to error. This is due to the fact that the noise making up the wide band level may represent either a single tone or a continuous

spectrum. The difference between these two possibilities is the maximum error which can occur. It is equal to ten times the logarithm of the ratio of the wider bandwidth to the narrow bandwidth and is expressed in decibels.

When the Detection Differential is calculated from any analysis of greater than 50 cps bandwidth, it is subject to these errors which can be corrected by the addition of Recognition Differentials which represent the amount by which any given spectrum varies between a single tone and a continuous spectrum. Fig. 13 is an example of the calculation of the Detection Differential from the spectrum of the under-

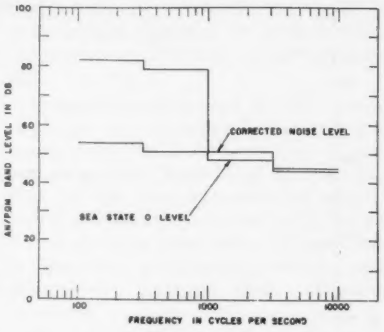


Figure 13
Calculation of Detection Differential from Measurements Made in AN/PQM Bands

Frequency CPS	Measured Level-DB	Background Level-DB	Corrected Level-DB	Sea State 0 Level-DB	Detection Differential
100-316	82	60	82	54	28
316-1000	79	57	79	51	28
1000-3160	56	54	51	48	3
3160-10000	51	50	45	44	1

$$DD\ (PQM)_{max} = 28\ db$$

water noise produced by a typical item of submarine auxiliary machinery. This calculation is made from frequency analyses in bands of 100-316, 316-1000, 1000-3160 and 3160-10,000 cps. The background noise for which the correction is shown is the ambient noise of sea state zero.

When determined from a 50 cycle bandwidth analysis, the Detection Differential may be used to determine directly the detectable range of a given noise of sea state zero. Under ideal sea conditions, taking into account only the spherical spreading loss, the detectable range is expressed by the formula:

$$r = 2^{DD/6} \times d$$

where d is the distance from the noise source at which the Detection Differential is determined. For a noise measurement made alongside a ship, d is assumed to be 5 yards. Curves showing the detectable range for various sea states for overside measurements and measurements made at 100 yards abeam are given in Fig. 14.

When determined from a 50 cycle bandwidth analysis over a frequency range to which a given passive sonar responds, and combined with a distance correction factor, the Detection Differential will give a reasonable measure of the radiated noise created by any given machine aboard ship.

When calculated over the band spread of a given active sonar, and combined with a directivity index and suitable distance factor, the Detection Differential can give a measure of the self noise due to any given machine.

CONCLUSION

Each sound which interferes with submarine sonar or which makes a submarine detectable by another vessel has its own peculiar character. That

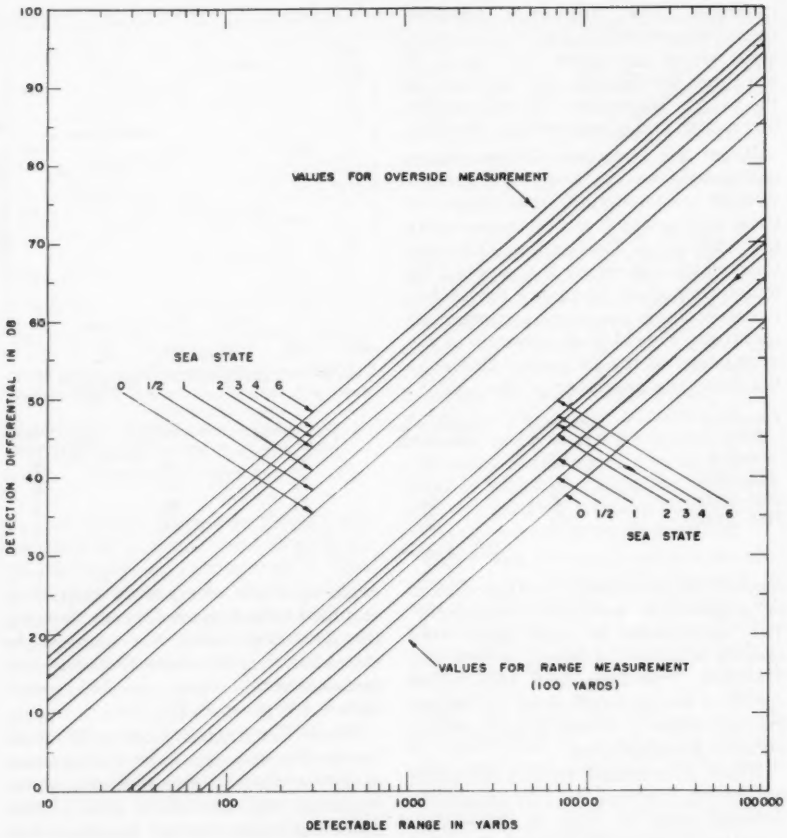


Figure 14
Curves for *Approximate* Determination of Detectable Range from Measurements Overside and at 100 Yards.

character is a function of its relative amplitude at various frequencies and of its variation with time. A sonar operator can be trained to recognize and identify sounds coming from another ship and to partially disregard noises emanating from his own vessel.

However, to the naval engineer who is attempting to eliminate interference in sonar and reduce the detectability of our own vessels, the problem is far more complicated. He must first be able to sort out and identify unwanted noise sources. He must understand which of these sources are important from an aural recognition standpoint. He must then devise means to reduce the noise emanating from

each source either by elimination, redesign, isolation, or relocation. Such detailed analysis of the problem is not feasible by listening alone, but must be made on the basis of detailed measurements and intelligent interpretation of those measurements.

Many tools are available to help in this work. Physiological and psychological research has helped to determine the response characteristics of humans to sound and thus to define the limits of the underwater noise problem in terms of aural listening. Thousands of submarine noise measurements have been made and are available for reference. Specialized instrumentation has been devised to probe deeply into the intricacies of the problem.

Unfortunately, the available data are seldom in a form which can be readily interpreted by the average engineer. The instrumentation seems extremely complicated and the information supplied by different types of instruments appears to conflict with subjective reports from ships' personnel.

It cannot be expected that the naval engineer can completely master the complicated field of acoustics. However, a general knowledge of the subject, of the characteristics of sound, and of the instruments employed to define these characteristics are essential to produce submarines which can adequately perform their assigned missions.

This article has attempted to outline briefly a few of the pertinent factors which relate acoustics to naval engineering. It is hoped that this partial treatment of one phase of the subject will stimulate further interest in this field which is so vital to the successful design and construction of naval vessels.

FREE-PISTON GAS-TURBINE PRIME MOVERS

ACKNOWLEDGEMENT

This article, which reviews the basic principles of free-piston gas-turbine prime movers, was published in the March 1955 issue of *Mechanical Engineering*. It is based on a paper presented at the Annual Meeting of the American Society of Mechanical Engineers, 28 November-3 December, 1954, in New York. The author is A. J. Ehrat of the Baldwin-Lima-Hamilton Corporation, Hamilton, Ohio.

Active American evaluation of free-piston gas-turbine prime movers has been continuing since before the end of World War II, although on a very limited scale in proportion to that of gas-turbine prime movers as a whole. During the early years the fundamental work was carried on through the efforts of the U. S. Navy Bureau of Ships and the author's company, but now at least two other firms are engaged in separate evaluations. There are also many other signs indicating a general spreading of interest.

WHY THE FREE-PISTON GAS TURBINE

The extent to which free-piston gas-turbine machinery will be applied cannot be predicted with accuracy at this time but it is the author's conviction that such equipment can take its place beside other successful prime movers. In comparison to diesel engines, free-piston machinery has inherent advantages of lower specific weight, lower maintenance costs, greater flexibility of arrangement and operation when used in multi-gasifier plants, and greater possibility of delivering large amounts of power to a single shaft.

Free-piston machinery has an efficiency advantage over small steam or simple-cycle gas-turbine plants. Also, in comparison to steam or complex-cycle gas-turbine plants, the hazard due to the escape of large quantities of high-temperature working fluid from large boiler or receiver-type volumes is virtually eliminated.

THERMODYNAMIC CYCLE

The basic elements of a free-piston-gasifier gas-turbine plant are a pair of single-stage reciprocating compressors, a two-stroke-cycle opposed-piston diesel cylinder which is coupled to the compressors, and a power

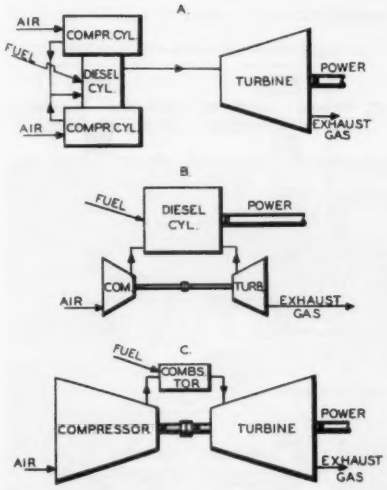


Fig. 1 Flow diagrams showing *(A)* free-piston gas turbine; *(B)* turbo-charged diesel engine; and *(C)* simple-cycle continuous-combustion gas turbine.

turbine. As shown by the schematic flow diagram, Fig. 1*(A)*, air drawn from the atmosphere by the compressors is compressed and delivered to the diesel cylinder, where it is compressed further. Fuel is then injected and burned, and the combustion products are expanded partially in the diesel cylinder and the rest of the way in the turbine. The power developed by the diesel cylinder drives the compressors, and all of the power developed by the turbine is delivered to the output shaft.

Fundamentally, the free-piston gas-turbine plant employs the same ideal thermodynamic cycle as do its better-known counterparts, the turbocharged diesel engine, Fig. 1*(B)*, and the simple-cycle continuous-combustion gas turbine, Fig. 1*(C)*. This cycle involves isentropic compression of air drawn from the atmosphere, constant-pressure combustion in this air, and isentropic expansion to atmospheric pressure. It is commonly known as the "constant-pressure" or Brayton cycle, Fig. 2.

Although all three of the prime movers operate on the same thermodynamic cycle, important differences result from the several methods by which the processes of that cycle are effected. The free-piston gasifier and the diesel engine both use an engine cylinder in which combustion is carried out intermittently, whereas the gas-turbine plant employs a chamber in which fuel is burned at a relatively steady rate. Therefore, because much higher maximum cycle pressures and temperatures are permissible in the diesel cylinders of the gasifier and of the diesel engine, the cycle efficiencies of the two are comparable. Furthermore, even though the maximum cycle temperature is higher, the temperature at the turbine inlet can be appreciably lower than that of the continuous-combustion gas turbine because a major portion of the expansion is carried out within an engine cylinder.

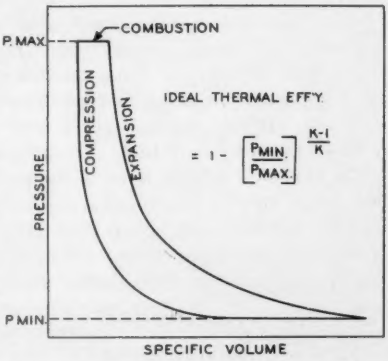


Fig. 2. The free-piston gas-turbine plant employs the constant-pressure or Brayton cycle.

GASIFIER CONFIGURATIONS

All existing free-piston gasifiers are symmetrically arranged to contain a central diesel cylinder flanked by a pair of compressor cylinders and by one or more pairs of bounce, or cushion, cylinders, Fig. 3. A composite diesel compressor-bounce piston assembly is housed within each half of the machine. By reason of the symmetry of the gasifier, at any particular instant both piston assemblies are displaced the same distance outward from the center line and the primary forces, accelerations, and velocities involved are equal and opposite in direction. In other words, the gasifier enjoys inherent dynamic balance, a trait which permits use of simple supports rather than a heavy foundation.

In spite of the design symmetry of the gasifier we may expect relatively small differences between the forces on the two piston assemblies, resulting from secondary variations in friction coefficients, valve action, and the like. Therefore the two pistons must be connected by a light mechanism which insures synchronization but which does not otherwise limit or control motion. This synchronizing mechanism may consist either of a pair of toothed racks operating on a common pinion gear, or of a system of linkages. It has another very important function in that it is used also to drive the fuel pump and to time the injection of fuel.

The gasifiers which have actually been built may be classified as one or the other of two basic types. They are either outward-compression machines, Fig. 3(*a*), or inward-compression machines, Fig. 3(*b*), depending on the direction in which the pistons move during the compressor-cylinder compression and discharge period. Each type, of course, can have a number of variations.

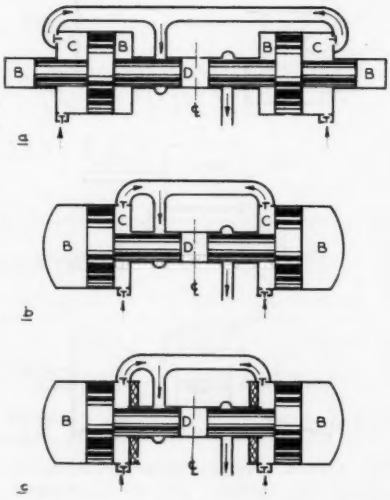


Fig. 3. Basic types of free-piston gasifiers: (*a*) Outward compression; (*b*) inward compression; (*c*) inward compression with adjustable compressor heads.

Fig. 3(*c*) depicts the inward-compression-type variation which involves the use of adjustable compressor heads. This is the machine which will be used to illustrate the principles of free-piston gasifiers.

MODEL DL GASIFIER

A more detailed sectional view of an inward-compression-type machine with adjustable compressor heads is shown in Fig. 4. It is the Baldwin-Lima-Hamilton model DL, a high-output unit developed for the U. S. Navy Bureau of Ships.

The gasifier actually under test has

an 8-in. diesel cylinder bore, a 22-in. compressor-and-bounce cylinder bore, and a nominal net stroke of 10¾ in. Its speed is of the order of magnitude of 1000 cycles per min. Nominally the machine is rated at 700 gas horsepower which implies an output, at the shaft of an 85 per cent efficient turbine, of

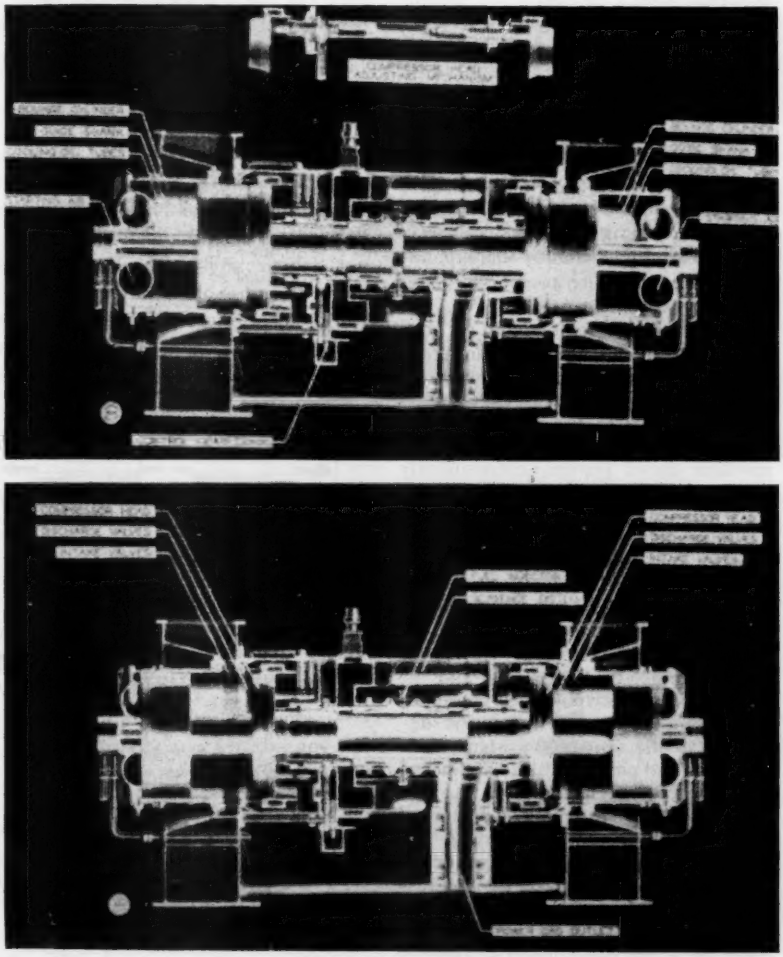


Fig. 4 BLH model DL gasifier, a high-output unit developed for the United States Navy Bureau of Ships.

around 600 hp. Actually it has already delivered as much as 830 gas horsepower, corresponding to 700 shp, during laboratory tests. The prototype machine has accumulated several thousand hours of operation, including a successful 500-hr official Navy endurance run.

The model DL free-piston gasifier has been operated only on distillate fuel oils. The satisfactory combustion in the gasifier of lower grades of fuel, such as the residuals, is entirely possible but has not yet been investigated because of the priority assigned to other work.

WORKING PRINCIPLES

The events during each cycle of the piston occur as follows, Fig. 5. The composite pistons are driven outward by the expanding combustion gases of the diesel cylinder and by the expanding air of the compressor-cylinder clearance volumes. Air is drawn into the compressor cylinders, and air contained within the bounce cylinders is compressed for storage of energy. Near the outer end of the stroke first the exhaust ports and then the scavenge ports are uncovered by their respective diesel pistons, consequently releasing the combustion products to the turbine and permitting fresh high-pressure air from the scavenge system to enter and purge the cylinder. The pistons are then returned inward by the energy stored in the bounce cylinders. The ports are closed off, the air charge thus trapped within the diesel cylinder is compressed, and the air previously drawn into the compressor cylinders is compressed and delivered to the scavenge system. Near the inner end point of stroke a metered quantity of fuel is injected into the diesel cylinder, causing repetition of the entire process.

The functioning of the free-piston machine is characterized by the fulfillment of two basic requirements. During each cycle of the pistons there must be a complete internal balance of work and during each stroke of the pistons there must be a complete balance of energy.

The first requirement, that the work shall balance, arises from the fact that the gasifier is only pneumatically linked to the load and therefore cannot deliver any external mechanical power. Because of this, all the indicated work of the diesel cylinder must be absorbed and balanced by the other components of the gasifier. Specifically,

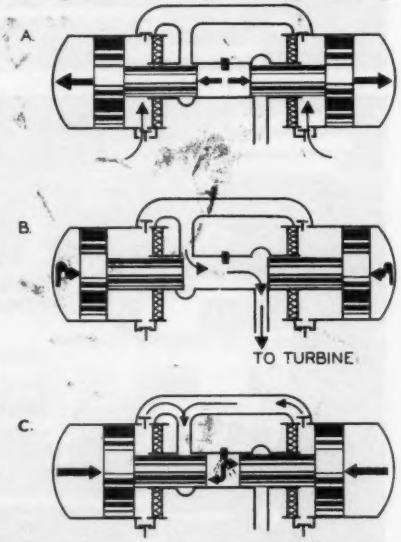


Fig. 5 Diagrams showing cycle of events: *(A)* Pistons driven outward; *(C)* outer end of stroke; *(C)* pistons returned inward.

the work of the diesel cylinder is balanced by the work of the compressor cylinders plus that of friction. This work balance is not a precarious one. On the contrary, within the broad operating range of the machine it is wholly automatic and is the basic means for adjusting power output.

The second requirement, that the energies shall balance, is attributable to the facts that the pistons reverse direction at each end point of stroke and that there are no rotating masses to carry kinetic energy over from one stroke to the next. Thus, during each stroke, all the energy delivered to the piston must have been absorbed from them by the time the extremity of stroke is attained. For example, during the outward travel of the pistons the expanding gases in the diesel cylinder and the expanding air in the compressor-cylinder clearance volumes transfer energy to the pistons, where it appears as kinetic energy. This energy subsequently is absorbed and balanced by the air in the bounce cylinders and by the comparatively small amount of friction.

Because of the energy-balance characteristic we may conclude that a free-piston machine contains no mechanical "flywheel effect." This is a unique advantage, for the machine can be brought up to speed swiftly after starting, can change load rapidly, and can be stopped instantly and safely in an emergency.

PRINCIPLE OF PISTON MOTION

The underlying principle of piston motion is much the same as that of a system composed of a mass oscillating between opposed springs. Although in the gasifier system the "springs" are pneumatic rather than mechanical, in both systems one spring projects the pistons against the other and the surplus energy released by the first subsequently is absorbed and balanced by the other. Because the point at which the stroke terminates and the motion reverses is the point where the balance of these two energies is complete, it may be shifted by alteration of the energy conditions through adjustment of the constants and working levels of the springs. As is true also of the mass of the spring-mass system, the gasifier pistons operate at a natural frequency which is a function of the reciprocating weight and of the force-versus-stroke characteristics of the springs.

Hence we observe that the motion of the pistons is rational even though not subjected to the inflexible restraint of the conventional crank-connecting rod mechanism, and that it is "free" only in the sense that it is free to be controlled through application of the work- and energy-balance principles. This freedom of control permits adjustment of stroke end points to suit the requirements of each operating situation.

ADJUSTING STROKE END POINTS

Adjustment of stroke end points obviously implies simultaneous adjustment of net stroke length and of piston displacement. Therefore it implies also a tendency to vary the air delivery of the compressors and the gas delivery and the gas-power output of the gasifier.

The outer end point is basically more effective for stroke-length adjustment than is the inner, because it can be varied over a much greater range. This

range, which is of the order of 20 per cent of the nominal piston stroke, falls between an end point so great that the minimum safe operating clearance from pistons to bounce-cylinder heads is encroached upon and one so small that the scavenge ports are choked beyond the limit of stable gasifier operation.

The adjustability of the inner end point is at least as significant as is that of the outer end point, because it is used to adjust the diesel-cylinder compression ratio. As a result, the diesel-cylinder compression and firing pressures may be controlled throughout the load range.

COMPRESSOR CLEARANCE VOLUME

Another variable which has particular influence on the operation and power output of a gasifier is the clearance volume of the compressors. The theory of reciprocating compressors tells us that for high volumetric efficiency this clearance should be as small in proportion to the displacement as possible. This requirement assumes added importance as the pressure ratio is increased.

On an outward-compression gasifier the outer end point of stroke governs the clearance volume. This relationship is reasonably satisfactory, the only objection being that there is a slight variation of the position of this point from cycle to cycle and as a result the linear clearance of the compressors cannot with safety be reduced quite as much as might be desired.

On an inward-compression gasifier the inner end point of stroke governs the clearance. This is an advantage because the inner end point is very steady. Unfortunately, the advantage gained is clouded by the fact that now compressor clearance is coupled also to diesel-cylinder compression ratio. For a machine of high specific output such an alliance is detrimental because the requirements of these two very important operating variables conflict. That is, as turbine-inlet pressure rises, on the one hand compressor clearance should decrease or should at least remain constant in order that highest volumetric efficiency may be maintained, but on the other hand diesel-cylinder clearance should increase in order that compression and firing pressures may remain at reasonable values. Whereas the first consideration demands a reduced or at least a constant inner end point of stroke, the second consideration demands an increased one. The opposing requirements can be compromised to a certain extent up to moderate turbine-inlet pressures, say, 50 psig, but at higher pressure the differences become too great for practical operation. Even at lower pressures, the necessary compromise tends to reduce the specific output as well as the operating flexibility of the gasifier.

On an inward-compression gasifier with adjustable compressor heads there is no need for such a compromise because the compressor clearance is basically divorced from the stroke end points. Aside from the fact that the steadiness of the inner end point permits continuous operation with a very small compressor linear clearance, which can be as little as 1 per cent of the net piston stroke, neither the inner nor the outer end point has a primary influence on the choice of compressor clearance. The resulting separation of compressor-clearance control from that of diesel-cylinder compression and firing pressures is especially advantageous because the machine can be operated under conditions which

are optimum as regards endurance, thermal efficiency, and specific power output.

Whereas the inward-compression machine with fixed compressor heads is limited as to maximum turbine-inlet pressure, that with adjustable heads can be operated at any pressure which it can physically withstand. In this respect the latter has an advantage in common with the outward-compression gasifier. An additional advantage is its capacity for operating at appreciably lower turbine-inlet pressures than can gasifiers of the other two designs. This means that the inward-compression gasifier with adjustable compressor heads can idle more easily against a turbine.

COMPATIBILITY OF GASIFIER AND TURBINE

The mechanical power delivered at the turbine shaft is the product of the adiabatic turbine-shaft efficiency and of the adiabatic gas power delivered by the gasifier. A gasifier, when operated as a separate entity, can furnish each desired value of gas power in the form of many different combinations of gas flow, gas pressure, and gas temperature. If the gasifier were operated against a turbine equipped with an infinitely variable nozzle area the combined plant could enjoy the complete operating flexibility of which the gasifier is capable. Such a goal would, however, require very complicated controls because each power requirement could be satisfied by an infinite number of gasifier and turbine operating conditions. Therefore it is likely that, even in the particular applications where a variable-nozzle-area turbine might be used, a definite schedule of area variation would be adopted.

In most instances, however, a turbine having a fixed nozzle area is all the free-piston plant requires. The actual value of this area is chosen at the outset of the application study and depends on the nature of the specified installation. Generally speaking, smaller nozzle areas permit the attainment of higher maximum powers because the gasifier can supply higher pressures and temperatures before it reaches its gas-flow limit. On the other hand, smaller nozzle areas also imply higher pressures and temperatures for each absolute value of part-load power and thus involve more severe operating conditions. Hence smaller nozzle areas would be chosen for applications requiring higher powers for short periods of time, as is true for many naval combatant vessels, and larger nozzle areas would be chosen for continuous full-power installations.

Once the nozzle area of a fixed-nozzle-area turbine has been chosen, the delivery characteristics of the gasifier must match the consumption characteristics of the turbine at all loads.

CONCLUSION

In conclusion it may be stated that the text of this paper forms only an introduction to the principles of free-piston gas-turbine prime movers. The principles themselves are neither fundamentally new nor, it is hoped, difficult to understand. It is necessary only that they be approached objectively. For additional, more specific, information concerning this interesting application the engineer is referred to the literature, some excellent examples of which are given in the references which follow.

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THE IONOPRINT AIRBORNE PRINTING SYSTEM

MIDSHIPMAN HAROLD L. MOONEY, U.S.N.A.

THE AUTHOR

began experimenting with electronics at the age of twelve. Obtained F.C.C. amateur license for station W2YSS at age fifteen. Entered Navy at outbreak of Korean conflict upon graduation from high school in 1950. Graduated from U. S. Naval Electronic Technicians School, Great Lakes, Illinois, in July 1951. Served fifteen months aboard U.S.S. *Amphion*, AR-13, in east coast area. Various other duty posts in Charleston. Completed enlistment prior to entering Academy in spring of 1953.

At the April meeting of IAS Southeastern Regional Conference at Virginia Polytechnic Institute, Midshipman Harold L. Mooney, Jr., third class, won first prize with his paper on "An Airborne High-Speed Electronic Printer." His technical paper was in competition with undergraduates of 12 other schools, including Geogia Tech, Alabama Poly, and the University of Maryland.

Many of the competitions in these contests are upper classmen majoring in aeronautical engineering. Midshipman Mooney has had no previous engineering training in college and was the only sophomore in this regional competition. IAS does not hold a national contest.

An atomic bomb has just been dropped on Washington, D.C.!! This may not be a pleasant thought, but it could have just happened. For our purposes though, let us suppose that this weapon has just been detonated over the heart of Washington.

From distant airbases planes from the Strategic Air Command will be winging their way to bomb targets deep within enemy territory. These operations will be coordinated from a central planning system. All local airfields will be under a tremendous work load trying to handle air evacuation of the wounded to hospitals in distant cities. At the same time, the incoming traffic will be trebled or quadrupled by planes flying in food, emergency equipment, clothing and blankets. The airwaves will be jammed by radio transmissions of all sorts from those requesting landing instructions to those asking what to do when they have run out of fuel. It might be this type of communication and coordination emergency that could tieup and cripple all efforts to bring relief to this devastated city. This type of organization panic and communication failure could mean life or death to thousands of people who might be depending upon the airlanes for survival as did the people of Berlin after World War II.

This may be citing one of the worst

possible tieups that *could* occur in the United States today. But we must remember . . . it *could* happen at this moment. There are many situations that might occur which would necessitate rapid and accurate communication such as the washing out of a small airfield by a rainstorm, a crash of a large airliner on the duty runway, or perhaps failure of the VHF voice radio system. Therefore, we *must* have a system that can provide us with reliability, accuracy of data, and speed of operation.

Let us then try to visualize such a system in one of our present day airliners or military aircraft. The use of voice radio could be seriously impaired by atmospheric conditions, or panic on the part of the control tower operator or pilot. An entire message could be misinterpreted by the pilot or misread by the tower operator and thus endanger the lives of many people by failure to perform a certain operation. Add to this the confusion caused by other planes trying to get into the voice circuit and you might end up with complete failure of the voice communication network. Granted that there are rules promulgated by the Federal Communications Commission and the C.A.A. Who is going to obey all of the rules when he is out of gas??

Radio Morse code is virtually out of the question. A trained radio operator would be necessary to operate the receiver. This would require added weight and expense. If the co-pilot were to be utilized for this purpose, he would have his hands bound by the limited speed of copy. Obtaining a Federal Communications Operator's license requires a code proficiency of from fifteen to twenty-five words per minute. The maximum that can be copied by an experienced operator with reasonable accuracy is twenty-five words per minute. Perfect accuracy is almost unheard of whenever weather conditions are bad. We would require a receiving speed of from eighty to one hundred and fifty words per minute. Thus, we have eliminated the Morse code operator.

We are now left with two alternatives: teletype or television. Television can be ruled out due to the unreliability of electronic components in such a complicated device. Also, the power and weight requirements of an airborne television system would far outweigh its practical advantage in cost and space.

Teletype must be the answer. It meets the requirements of accuracy and speed of communication if it were not for one major factor—weight. A standard Navy teletype system weighs close to three thousand pounds complete. Even more discouraging is that it requires close to 200 cubic feet of space for proper mounting and ventilation. Today, with the requirements of aircraft engines and fuels computed to such close tolerances for specific jobs, each pound of excess equipment aboard military aircraft will amount to an overall increase of $1500 in the cost of the plane to compensate for extra wiring, fuel consumption, load requirements for the engines, and increased safety factors. At this rate, a standard teletype unit would increase the initial investment and operating costs *only* four million dollars. Naturally, we have not included the cost of the unit in this figure either . . . . . this alone could be in the six digit bracket. With this cost summarized, the overall cost of such a system would be far more than that of the original plane, pilot, crew, and enough gasoline to fly it to the moon and back.

Obviously such a unit would be virtually worthless when costs are added up. But what if we had a system that was lightweight, compact and cheap? What use could we put such a unit to? How could we utilize a unit that would draw minute amounts of power from the plane's generator sys-

tem and might be able to do many jobs? The answer is very simple. There is no limit to the applications of such a unit!

Such a system could have a small printer located in the cockpit of the plane to feed information to the pilot regarding weather conditions, conditions of various runways in the area, direction and velocity and direction of the wind in his operating area. All of this information could be relayed to the plane automatically by just tuning this system in on a certain channel or by calling into control and requesting. The regular VHF channel would then be free from this type of traffic. The control tower operator could then concentrate on other jobs and leave this to be done automatically by a small data assimilating and transmitting device. A small unit of this type could be rigged in conjunction with the regular air search radar at the control station and could easily relay to the plane automatically the speed and course of any planes that might be in the same operating altitude and thereby remove much of the danger of mid-air collision. A bell or light could be wired in parallel with the printer unit to give the pilot audible or visual warning of a possible collision vector.

An even more valuable application of airborne teletype would be to couple this unit *directly* with the airsearch radar via an isolated VHF channel. The pilot could call into the tower and request progressive ranging. The control tower operator would then have the radar ranging system locked on the plane. The radar range rate pulse that would normally be connected to the radar console would then be relayed over this VHF channel to the plane. Here the teletype unit could incorporate one of its frequency shift networks to power a continuous range indicator on the pilot's instrument board. Under conditions where he would normally have a difficult time judging his range from the airfield, he could now have a continuous reading of his flying distance within one hundred years. Once again, we would be keeping the VHF channel open for emergency traffic and would be cutting down on the number of jobs to be handled by control tower personnel.

Since we now have a means of measuring relative distance, it would be no problem to plot this against time and figure average ground or airspeed of the plane approaching the field. If this data were relayed automatically back to the unit aboard the plane, the pilot would have more valuable information to enable him to use a minimum of fuel in making his approach, keep him from overshooting the runway or perhaps eliminate the possibility of a slow speed stall.

Now that we have seen many valuable applications of a small lightweight teletype unit the only major problem that remains is its design. One of the greatest problems that must be considered is the elimination of weight. Almost all teletype systems that are now in use are constructed on the regular typewriter principle. A ribbon moves over the surface of the paper and the keys are electrically actuated to strike in one position at all times. Mechanical equipment is required to move the carriage across the base in front of the keys. Electrical and mechanical equipment is required to make sure that the unit will operate the carriage at the proper speed for each sending speed. Up until now, teletype designers have not been concerned with the weight of such units. They have always been used in such locations that a few thousand pounds one way or another would make little difference. These units have been constructed primarily of steel to assure strength and shock resistance. This is particularly so in Naval installations where gunfire and depth-charge attacks cause severe vibration and necessitate reinforcement of all delicate

portions of the unit. Every bit of reinforcement means additional weight and space. The majority of teletype units now in operation are constructed of medium carbon steel alloys. Thus by incorporating aluminum or magnesium in the construction of our unit, we would be able to eliminate over half the weight.

Within the past few years, rapid development has been made in a new type of vacuum tube called the transistor. Actually it is not a vacuum tube but more of a device like the old time crystal set. It weighs less than one twentieth the weight of a vacuum tube of the same circuit specifications and is as small as the button on your doorbell. What is even more important is that it uses seventy percent less power. This would mean that much of the weight carried by an airplane in generators to power radio equipment could be reduced by the incorporation of such tubes. Furthermore, they are not subject to the failure found in regular electronic tubes due to shock and vibration. Hence, by the use of aluminum and transistors in our system, we can increase the reliability three fold and decrease the weight almost ninety per cent. We now have our unit extremely reliable and accurate and weighing three hundred pounds. This is still too much for practical application.

The greatest amount of weight left to be reduced will be found in the printer unit of a standard teletype system. Therefore, it will be necessary to redesign the printing section of the unit. If we could eliminate the moving carriage and the weight of equipment such as the electric motors used for the drive system and the cams and timing devices, we would be able to cut down a substantial amount of weight. With most of the weight of this unit removed, our printer can be reduced to approximately one-hundred pounds, or down to one thirtieth of what the original standard system weighed.

Since we have eliminated the printer that we were going to use, it is necessary to find some system to print the information that would be transmitted to our unit receivers.

Several years ago while in New York City, I noticed an unusual sign in the heart of Times Square. This sign had a whole series of lights arranged in a grid fashion about fifty feet on a side. By turning on certain lights, they were able to produce pictures of animated cartoons. Then I wondered . . . . . why couldn't this same principle be used to print data in a lightweight system for aircraft. The applications for such a unit at the time seemed to be many. The only catch was . . . how to use these lights to make a permanent record too.

The first step to be considered was that the use of filament type lamps would be out of the question because of the time necessary for heating. Power requirements would be high and the heat generated by such lamps would have to be dissipated somehow. *What would be wrong with using vapor lamps?* They light instantly when voltage pulses are applied. If these bulbs were arranged in the shape of letters and their light was recorded upon a sensitive emulsion, it would be possible to have a permanent and wholly accurate record of all information transmitted to it.

It is this system I call the IONOPRINT high speed printer . . . its name coming from the need to ionize these gases to produce light. Where the conventional teletype system is limited to 120 words per minute the IONOPRINT UNIT will print as many as 450 words per minute . . . or even higher if a more sensitive emulsion were to be developed. Perhaps figure No. 1, the block diagram, and the explanation of the individual stages will make the overall system clearer.

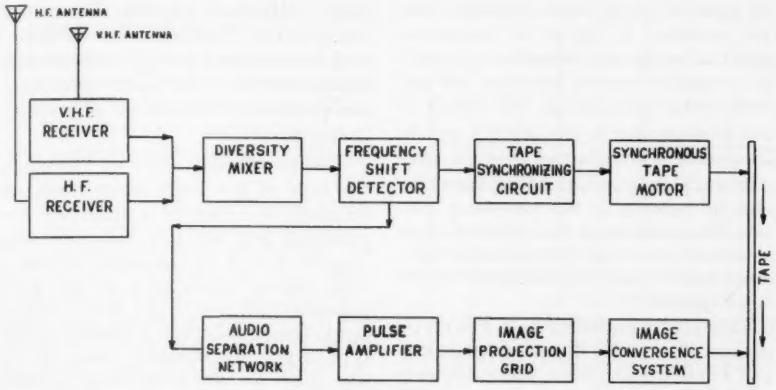


Figure 1
Simplified Block Diagram of the Ionoprint System

PURPOSE AND OPERATION OF INDIVIDUAL CIRCUITS

Diversity Mixer

The purpose of this unit is to mix the two separate radio frequency channels and thereby provide a stable incoming frequency for the detection of the audio frequency. By having the incoming frequencies differing by such a wide spectrum, it will be possible to combine the long range of the Low Frequency transmission with the interference-free Very High Frequencies which, in the event of possible jamming, may be directively beamed to the receiver by a radar tracking and directing system. It is not absolutely necessary to use this combination though, as two VHF, HF, or LF channels may be used depending upon whether or not the incoming frequencies originate from an aircraft carrier operating in enemy waters where the detection of uni-directional signals is possible. Since it is far simpler to beam VHF signals than the lower frequencies, this will provide an optimum in security and radio silence measures. With the recent growth of Infrared or NANCY gear, it would be little trouble to couple an Infrared receiver in conjunction with the regular spectrum receiver and thereby eliminate the need for radio silence.

Frequency Shift Detector

This unit is designed to separate the radio or Infrared frequencies from the audio frequencies superimposed upon them. This is accomplished by a vacuum tube or a series of tubes that will act as a blocking system to all signals of the radio frequency spectrum and pass only those audio signals necessary to operate the Audio Separation networks. These audio frequencies can then be passed on and amplified to proper operating voltage if necessary. If radio frequencies were present in the Audio Separation Network, they would cause an unbalance of impedence in the network and produce erratic firing in the image projection grid.

Audio Separation Network

Each character or cipher (the word cipher is used because, if necessary, it would be possible with this system to transmit and print diagrams or rough pictures depending upon the number

of separate pulse light systems used, i.e., number of lights in projection grid) is assigned a definite frequency at a regular interval between 400 and 5000 cycles per second. The object of the whole system is that when a certain frequency is transmitted over the carrier wave, the corresponding character will be printed at the receiving section. The purpose of this network then is to make sure that this certain frequency prints only the character which it is supposed to.

This is accomplished by the incorporation of a series of audio band-pass and band-reject filters. Since the rate of printing will be in excess of 100 words per minute, it is necessary to have a completely electronic switching system for the different audio frequencies. For example, if the transmitter is operating at a speed so as to transmit one hundred words per minute, there will be only 120 milliseconds separation between the character pulses. At this speed the use of relays is impractical. As the speeds increase, it becomes impossible to use relays and electronic switching and separation becomes a necessity.

A whole series of frequencies are transmitted at the predetermined pulse interval. Each of these frequencies is fed into AUDIO SEPARATION NETWORK. Here they will cause resonance in the band pass filter of the same audio frequency. This will permit only this band of frequencies to pass. If by any chance there are spurious audio signals intermixed with this frequency (i.e., of a different frequency) they will be rejected immediately by the band reject filter. This will permit only the desired signal to actuate the image projection grid.

Pulse Amplifier

After the separate audio pulses have been separated from the undesired frequency groups, they are little more than low voltage pulses capable of little peak voltage or current. To operate the IMAGE PROJECTION GRID, it will be necessary to have a reasonably high potential, as the GRID operates on the principle of ionization of pressurized gases.

This potential is directly related to the type of gas used. Since neon can be made to ionize at a relatively low potential but has low light intensity output, it will be necessary to use a light source requiring a greater peak potential but emitting a more intense light. This is mandatory because of the type of printing emulsion intended for use. With the development of a more sensitive emulsion, this intensity may be reduced, and consequently the size of the IMAGE PROJECTION GRID may be reduced. Eventually, with ultra-sensitive emulsions, it will be possible to eliminate the pulse amplifier stage completely.

Image Projection Grid

This unit is the last step in the conversion of an audio signal to a visual recorded signal. Basically it consists of a grid mounted in front of a system of converging lenses. On the grid are eighty-one gaseous vapor lamps that are individually connected to the output of the PULSE AMPLIFIER system. Since it would be electronically inadvisable to have a separate lamp for each of the characters being fed from the pulse amplifier, many of these lamps are arranged in parallel to allow a whole series of lamps to light from one pulse from one PULSE AMPLIFIER and thus conserve tubes and space. For instance, in the letters: B, C, D. E, F, G, etc., it is necessary to use the light combination 1(A-I) . . . (see grid drawing for explanation of this code) which is a vertical series of lights on the extreme left hand side of the grid.

If we did not want to conserve space and weight, we could arrange to have a separate wire and circuit for this

group of lamps each time one of these letters is to be printed. To eliminate this, the character shapes are set up in a manner such that this combination of lights could be used in many different characters and thereby do away with a considerable bit of wiring and weight.

There are some lamps allowed in the grid that are of no immediate use as far as character printing is concerned. The way the letters are set up, these lamps would not be in the circuit. However, they could easily be connected in the event that it were necessary to change the shape of the different characters or incorporate the image or diagram transmission system. In this event, plug-in pulse amplifiers and audio separation networks could be added to the unit to use these lamps. The *primary* matter concerning us now is the use of this system to print word text . . . and to use a system that incorporates a minimum amount of space and weight.

This is effectively accomplished by the use of plug-in units and by designing the system so that future improvements may be added. If in the future a code system is devised that requires but a few characters or symbols, it will be possible to remove the unneeded stages leaving only the necessary ones to do the job. Conversely, if the stages presently available are found to be insufficient, it will be a simple matter to add a few more stages until the necessary number of lights can be made to operate and print their image.

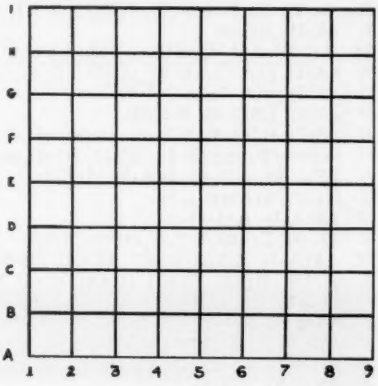


Figure 2
Image Projection Grid—Pulse Light at Each Grid Line Intersection.

In any event, it will be possible to lengthen the span of the lamps by rotating them in their sockets from the most used to the unused positions. The unused lamps can also be used for a spare parts (lamps) storage in case of breakage or failure of one of the regular lamps that is continually used. Figure 2 and Table I illustrate the lamps used for each letter character.

TABLE I

Key to Letter Designation in the Image Projection Grid

LETTER:	KEY:
A	I(1-9), 9(A-I), (A-I), E(1-9)
B	(A-I), I(1-8), 9(F,G,H,), 9(B,C,D,), E(1-8)
C	(A-I), I(1-9), A(1-9)
D	(A-I), A(1-8), I(1-8), 9(B,H)
E	(A-I), I(1-9), A(1-9), E(1-7)
F	(A-I), I(1-9), E(1-7)
G	(A-I), I,A(1-8), D(7-9), 8(A-D)
H	(A-I), 9(A-I), E(1-9)
I	5(A-I)
J	A(3-7), 7(A-I), (B-3)

K (A-I), (E-2), 3(D,F), 4(C,G,), 5(B,H), 6(A,I)
L (A-I), A(1-9)
M A(1-9), 9(A-I), H(2,8), J(3,7), F(4,6), (E-5)
N (A-I), 9(A-I), (B-8), (C-7), (D-6), (E-5), (F-4), (G-3), (H-2)
O (A-I), 9(A-I), A,I,(1-9)
P (A-I), I,E(1-9), 9(E-H)
Q (B-I), B,I(1-8), 8(B-I), (A-9), (C-7)
R same as P, add (B-8), (C-7), (D-6) and (A-9)
S I,E(1-9), A(1-9), 9(A-E), 1(E-I)
T I(1-9), 5(A-I)
U 1,9(A-I), A(1-9)
V (A-5), C(4,6), E(7,3), G(2,8), I(1,9)
W 1,9(A-I), B(2,8), C(3,7), D(4,6), (E-5)
X A(1,9), B(2,8), C(3,7), D(4,6), (E-5), F(4,6), G(3,7), H(2,8), I(1,9)
Y 5(A-E) F(4,6), G(3,7), H(2,8), I(1,9)
Z A,I(1-9), E(3-7), (B-2), (C-3), (D-4), (F-6), (G-7), (H-8), (I-9)
1 A(3-7), 5(A-I), (G-3), (H-4)
2 A,I(1-9), (B-2), (C-3), (D-4), (E-5), (F-6), (G-7), (H-8), (I-9)
3 A,I(1-9), E(4-9)
4 C(2-9), 8(A-I), (D-3), (E-4), (F-5), (G-6), (H-7), (I-8)
5 I,G(2-9), A(1-9), 9(A-G), 1(G-I)
6 A,I(1-9), 1(A-I), 9(A-D), D(1-9)
7 I(1-9), (A-1), (B-2), (C-3), (D-4), (E-5), (F-6), (G-7), (H-8), (I-9)
8 1,9(A-I), A,I(1-9), E(1-9)
9 9(A-I), 1(F-H), I,F(1-9)
ϕ 1(A-H), I(2-9), A(1-8), 9(B-I), (B-2), (C-3), (D-4), (E-5), (F-6), (G-7), (H-8), (I-9)

Image Convergence System

The size of the letter image at the IMAGE PROJECTION GRID is much too large to be transferred to a sensitized tape directly. It must be reduced to the size of a regular typewritten character. This is readily done by a series of convex reducing lenses. The printed character to be transferred to the sensitized tape should be 4mm on a side. This is a bit larger than the regular typewritten character but it will facilitate less eye strain on the part of the pilot. The Image Projection Grid is placed parallel to the first image convergence lens. When a particular character is pulsed by the Pulse Amplifier, the necessary vapor lamps ionize and the impression of each appears as a tiny dot on the sensitized paper after being condensed. (see Figure 3). This is shown in detail at the top of the diagram. If the pulse width of the character is lengthened, the period of ionization of the lamps will be greater and tape will tend to blur the dots (as occurs when you move your camera while taking an exposure with a slow shutter). The dots on the tape will then be blurred together and appear as a single line. The vertical lines on the tape may also be blurred into a single line by causing the tape to oscillate laterally a slight bit.

Tape Synchronizing Circuit

Thus far, we have not discussed the manner in which the sensitized tape will be made to run at the correct speed for each individual sending speed of the transmitter unit. We have seen that it is impossible for a teletype operator to key his unit at speeds up to four or five hundred words per minute. A means of overcoming this is to use the perforated tape transmission system or have the data recorded on a magnetic drum at slow speed and have the angular velocity of the drum varied in proportion to the speed of transmis-

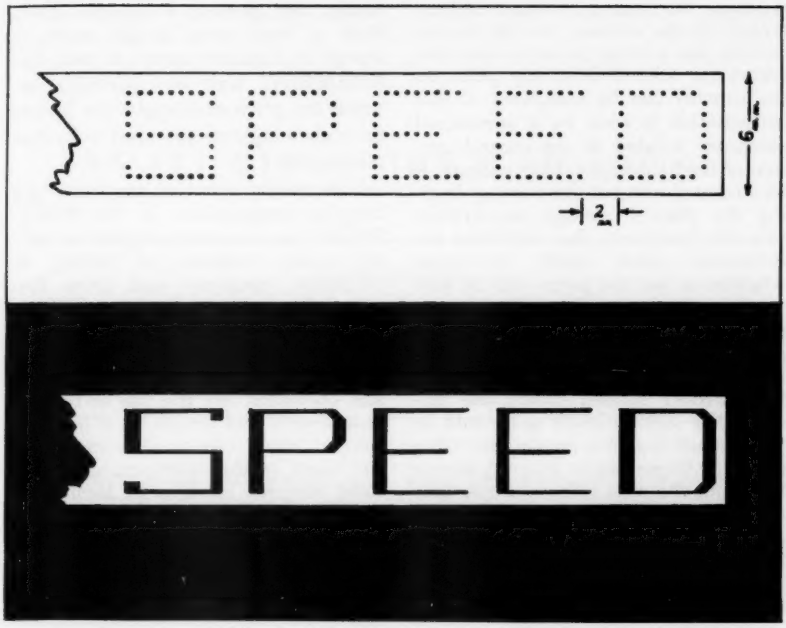


Figure 3

sion. This data is then applied to the vacuum keying system of our frequency shift transmitter. If we apply a synchronization series of pulses to our memory system, the faster the data are keyed, the greater will be the repetition rate of the pulses. Located in the IONOPRINT unit is a small multivibrator. The synchronization pulses will be filtered out of the Frequency Shift Detector and directed to the Tape Synchronization unit. An output frequency from the multi-vibrator dependent upon the pulse repitition rate input will be fed into the synchronous motor used to drive the tape at its operating speed. The speed of such a motor will be dependent upon the frequency of the multi-vibrator. In this manner, the tape can be accurately synchronized for any probable speed of transmission.

CONCLUSION

The IONOPRINT system need not be limited to recording data or as an aid to flying. Its greatest value might lie as a safety device aboard commercial airliners or military aircraft. Many of the accidents in the past have been by the planes' innability to attain sufficient airspeed at takeoff. This has been due in part to overloading, improper fuel mixture, overheating, poor visibility and poor judgement on the part of the pilot. Since the lift due to

windage, the mass of the plane and the length of the runway are all known factors, the average acceleration necessary for takeoff from any point on the runway can be computed. If this computation is done by a permanent computor located at the control station of the field and a radar antenna is located at the end of the runway feeding the plane's average acceleration into the computor, the combined information could easily determine whether or not the plane will be able to clear the ground in the runway space that it has left. If the computed point of takeoff lies beyond the runway, radio signals can be relayed to the IONOPRINT system aboard the aircraft. The IONOPRINT unit could be constructed in such a manner as to ring a bell or trigger some warning device to permit the pilot to have ample time to reverse the props and apply the brakes and prevent a possible crash. With a little extra design work, it would be a simple matter to have the IONOPRINT unit *automatically* reverse the pitch and apply the brakes through a lightweight electro-hydraulic system.

It is readily seen that the seemingly limitless applications of the IONOPRINT system make it capable of saving many millions of dollars in efficiency, property and lives that otherwise would be lost. It would enable civilian and military aircraft to meet the increased requirements for speed and certainty in communication and operation. By the use of reliable components and simplicity of installation, it could effectively compensate for many of the shortcomings of the most complex machine of them all—MAN!

EVALUATING SHOCK MOUNTS

ACKNOWLEDGEMENT

This article was written by J. P. Walsh and R. E. Blake of the Mechanics Division, Naval Research Laboratory, and was presented at the Annual Meeting of the American Society of Mechanical Engineers on 28 November-3 December, 1954, in New York. It was published in the March 1955 issue of *Mechanical Engineering*.

Shock mounts are flexible couplings used to attach equipments to structures which may be subjected to shock motions. They generally are much stiffer than vibration isolators. The greater stiffness is required because clearances for the equipments are not available for the large displacements which would result if the more flexible system were used. Therefore the shock mounts will aggravate much of the lower-frequency components of the steady-state vibrations and of the shock motions.

At present the principal criterion as to whether or not a shock mount is satisfactory for military use is that the equipment it supports withstand specified "shock" tests.

The objective of this work is to devise a procedure for determining those shock-mount characteristics which would (*a*) permit the evaluation of the relative effectiveness of different types of mounts, (*b*) provide a means of defining when a mount is acceptable, and (*c*) provide information useful to the designer of equipments to be supported by the mounts. This paper describes the methods developed with these objectives in view and presents some results for a few typical shock mounts.

CRITERION OF PERFORMANCE

A shock mount should be judged by its ability to protect an equipment from shock damage. It may be evaluated relative to a rigid support system, or in comparison with the effectiveness of other mounts. Two difficulties are immediately apparent if no limitations are placed on the nature of the equipment and on the nature and intensity of the shock motions:

1. The probability of damage to an equipment is greatly dependent on whether elements in the equipment might be strongly excited by frequency components passed, or introduced, by the mounting arrangement. These particular frequency components may have very little potential for damaging other equipment having differential natural frequencies of vibration. It

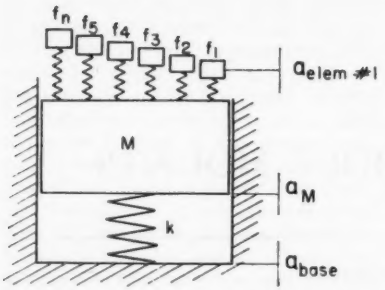


Figure 1. Rigid equipment M, shock-mounted by spring k, and containing elements having natural frequencies f_1 through f_n. Accelerations of parts are denoted by a with suitable subscripts.

will, therefore, be necessary to employ an idealized type of equipment for determining the effectiveness of a mount.

This equipment, shown in Fig. 1, is assumed to be rigid and contains a large number of single-degree-of-freedom systems which represent vulnerable parts of the equipment. The natural frequencies of these simple systems collectively cover the frequency spectrum of interest. All motion is along one axis and the masses of the elements are assumed to be sufficiently small as not to affect the motion of the equipment. The elastic stress which each of these elements must be able to withstand without failure is assumed to be proportional to the maximum force that acts on the mass of the element considered. This maximum force is proportional to the maximum deflection of the spring of the element, and to the maximum acceleration of the mass. Values of acceleration generally will be different for elements and, of course, will be different from the acceleration experienced by either the equipment or the mount. A quantitative expression for the effectiveness of a mount, for a given shock motion, is then expressed in terms of the accelerations experienced by these elements. The effectiveness is a specific value for each different element and is represented as a curve, Fig. 6 or 9, where the ordinate represents a measure of effectiveness and the abscissa represents the natural frequencies of the elements.

It is customary[1] to plot the maximum acceleration experienced by the element masses as this measure of effectiveness. This curve is frequently called the "shock spectrum" and the values are ordinarily labeled "equivalent static acceleration."

2. Whether one shock mount may be judged better than another mount depends upon the nature and intensity of the shock motion. It is, therefore, necessary to standardize on one or more types of shock motion and to use these consistently when comparisons are made.

The Navy logically would standardize on shock motions such as may occur on board ship. However, they vary from ship to ship and from one location to another on the same ship. But the shock motion delivered by the Navy shock machines is, in a sense, a standard shock. It is used to determine the suitability of equipment for naval use.

Although of interest and importance, the practical validity of the motions of shock machines does not affect the present investigation. This procedure has the advantage of flexibility. If the shock machines should be changed or modified, it would follow that the mounts would be examined on the modified machines.

Other types of tests were performed

[1] "The Equivalent Static Accelerations of Shock Motions," by J. P. Walsh and R. E. Blake, Proceedings of the Society for Experimental Stress Analysis, vol. 6, 1949, pp. 150-158.

to determine the static and dynamic physical properties of shock isolators. These include the determination of (*a*) static load-deflection curves, (*b*) dynamic load-deflection curves and energy loss by impact tests, and (*c*) dynamic elastic modulus by vibration tests.

DESCRIPTION OF IMPACT TEST APPARATUS

A hammer weighing either 100 or 175 lbs. was dropped between guides so as to strike the central section of a shock mount. Strain gages attached to a cylindrical element between the hammer striking surface and the main body of the hammer allowed force measurements to be made. A resist-

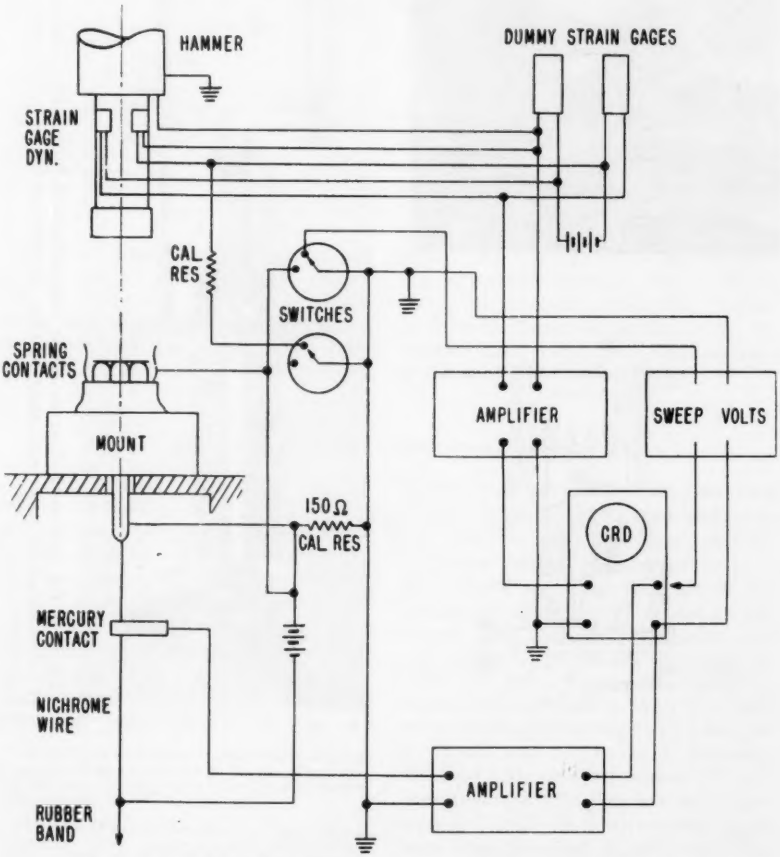


Figure 2. Schematic diagram of equipment for hammer-drop test

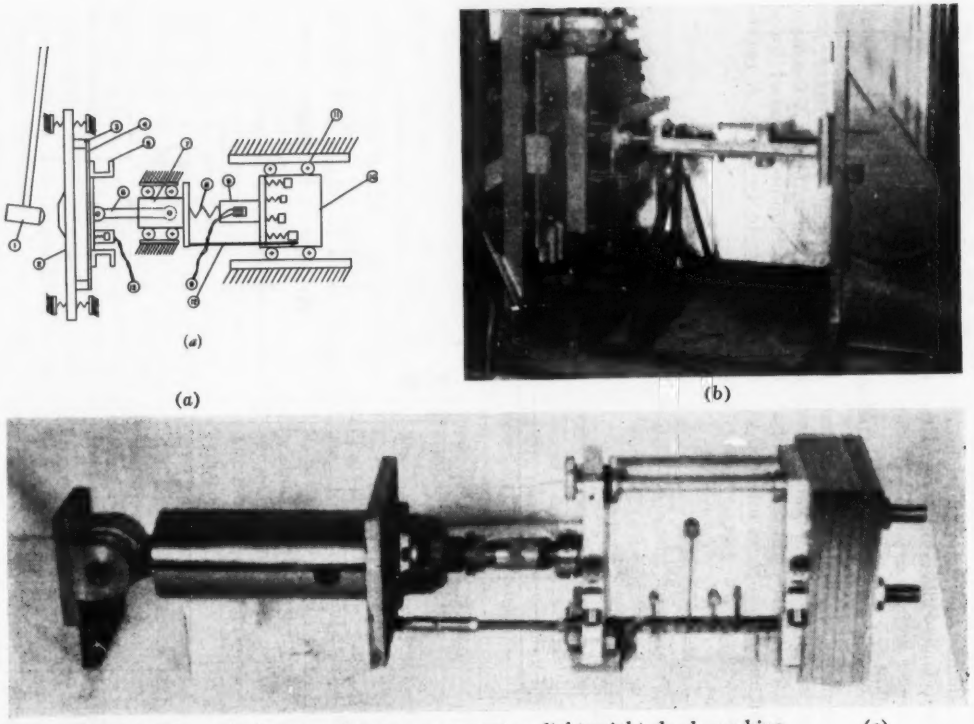
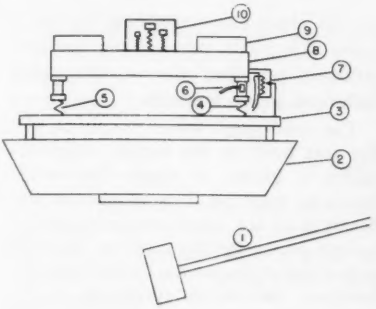
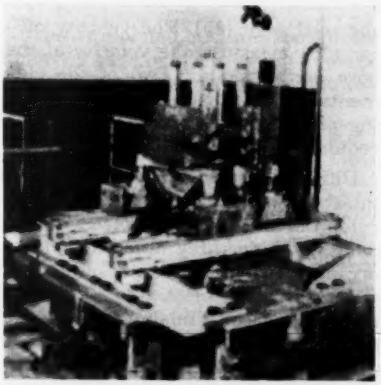


Figure 3. (*a*) Schematic diagram of test arrangement on lightweight shock machine. (*b*) Photograph of setup equivalent to (*a*). (*c*) Photograph of mount and idealized equipment.



(*a*)

(1) hammer
(2) anvil table
(3) mounting channels
(4) test shock mount
(5) auxiliary shock mounts identical with (4)
(6) calibrated dynamometer
(7) slide wire and maximum displacement gage
(8) load platform
(9) auxiliary weights
(10) reed gage



(*b*)

Figure 4. (*a*) Schematic diagram of test arrangement on medium-weight shock machine. (*b*) Photograph of setup equivalent to (*a*).

ance wire was attached at one end to the central section of the mount and was kept taut by a rubber band at its opposite end. The wire passed through a fixed-position mercury contact, which allowed the displacement across the mount to be determined. The schematic diagram, Fig. 2, illustrates the principle of this test. Curves of dynamic force versus deflection, or versus time, were recorded oscillographically by this equipment.

MOUNT TEST ON LIGHTWEIGHT SHOCK MACHINE

A description of the Navy light and medium-weight shock machines is given elsewhere.[2] Fig. 3 (*a*) is a schematic diagram of the lightweight shock-mount test apparatus. Figs. 3 (*b* and *c*) are photographs of the actual apparatus. On the diagram, (1) is the shock-machine hammer which strikes the anvil plate (2); (3) and (4) represent the spacers and ½-in. mounting plate shown in Fig. 4.[3] These items are parts of the standard machine. The mounting plate stiffeners (5) will be described later. A rod-crosshead system (6) is used to provide linear motion at the base of the mount (8) under test. The load on the mount (10) consists of a special reed gage and enough auxiliary weights to reach the desired mass. The range of loads available is 20 to 75 lb. The load is constrained to straight-line motion by tracks and rollers (11). Between the load (10) and the mount (8) is a dynamometer (9). This is a short column to which SR-4 wire strain gages have been attached. A slide-wire displacement gage (12) is integral with

[2] "Some Characteristics of Navy 'High Impact' Type Shock Machine," by I. Vigness, Proceedings of the Society for Experimental Stress Analysis, vol. 5, 1947, pp. 101-110.

[3] "Interim Military Specification Tests; Shock, Vibration and Inclination," MIL-T-17113 (Ships), July 25, 1952, BuShips, Navy Department, Washington 25, D. C.

the reed gage (10). The velocity meter (13) measures the velocity at the base of the mount. Two of the elements of this apparatus (the mounting-plate stiffeners and the reed gage) require further explanation.

Different transient frequencies in the shock motion are obtained by varying the stiffness of the plate. This is done by using three different bolting arrangements to fasten stiffening channels to the plate. These are (1) two bolts spaced 2 in. on either side of the vertical center line, (2) the first two bolts plus two more spaced 6 in. on either side of the center line, and (3) these four bolts plus two additional bolts spaced 12 in. on either side of the center line. These last bolts are at the channels supporting the plate from the anvil plate. This variable stiffness provides three essentiallly different shock motions.

The reed gage, which forms part of the test load on the mount, approximates a group of single-degree-of-freedom systems. The maximum deflections of the reeds are recorded on waxed paper by the scribers attached to the tips of the reeds. From these deflections the shock spectrum of the motion of the load can be computed by the method described previously. It should be noticed that this load is extremely rigid compared with most equipments. The effect of this unusually high rigidity on the results is discussed later.

MOUNT TEST ON MEDIUM-WEIGHT SHOCK MACHINE

In the case of the lightweight machine it was convenient to design the apparatus on the basis of testing one mount at a time, but the arrangement of the medium-weight machine makes it more convenient to use four mounts to support the test load. Actually, only one mount is instrumented completely; that is, one of the spool-shaped pieces connecting the mount to the load has strain gages attached and the slide-wire displacement gage is located as close as possible to it.

Fig. 4 (*a*) is a schematic diagram of the test apparatus used with the medium-weight shock machine. Fig. 4(b) is a photograph of the apparatus and test mount on the shock machine. The essentials of this apparatus are identical with the lightweight machine apparatus, and that discussion will apply except for the method of obtaining different stiffnesses of the mounting adapter.

For the purpose at hand it is required that the vibrations at the base of the mount be controlled enough to insure that if a frequency coincidence should exist during one test it would not exist on other tests of the same investigation. Since the load platform of the test apparatus has two possible center distances—16 in. and 24 in.—this can be accomplished by selecting three combinations of center distance and channel supports. The combinations which have been used are the following:

(*a*) Test-load center distance = 16 in., supporting channels for 24-in. center distance.

(*b*) Test-load center distance = 24 in., channels for 24-in. center distance.

(*c*) Test-load center distance = 24 in., channels for 24-in. center distance plus two car-building channels.

The instrumentation required for the tests on the lightweight and medium-weight shock machines is similar to that of the hammer-drop test. The force transmitted by a mount was determined from strain measurements on elements of known stiffness. Displacements were measured by means

of a slide-wire potentiometer arrangement. A velocity pickup indicated the motion of the shock machine, and a reed gage provided a shock spectrum of the equipment motion. Except for the reed gage, which was self-recording, all records were obtained by cathode-ray oscillographs.

TEST PROCEDURE

The method of taking data is apparent from the instrumentation but a few additional remarks are in order on the procedure of running a test of a shock mount.

Neither the directions of mount deflection nor mount orientation aboard ship are known at the time a test is being run, so one must consider all directions to be equally likely. However, to test a mount in many directions would be time-consuming and, fortunately, unnecessary. Usually, two directions are enough—along the axis of symmetry (most mounts have one) and perpendicular to it. These two directions will be referred to as axial and radial.

Hammer-drop and the lightweight machine tests have been designed to insure uniaxial deflection of the test mount. To test in the radial direction, a right-angle bracket is made and attached to the base of the cross-head. The mount is attached to the bracket so the motion of the load produces radial deflections of the mount.

The anvil table of the medium-weight machine moves substantially vertically. In order to test mounts radially they are rotated 90 deg so that the base of the mount, horizontal on the axial test, is now vertical. Brackets are made to secure the mount in this position with the base attached to the vertical leg of the bracket and the horizontal leg to the supporting channel.

There are cases when it appears to be of interest to conduct tests in which the direction of deflection is neither axial nor radial. In these cases special brackets are designed.

The independent variable in a test of a mount on the shock machines is the height through which the hammer falls before striking the anvil plate. Test runs are made using different stiffnesses of the mounting adapter with the height of hammer drop held constant. Each run with a given adapter stiffness produces a shock spectrum. These are averaged and the resulting spectrum, called the "averaged" or "composite" spectrum, is used to judge the mount.

RESULTS

In the course of developing this method of determining the characteristics of shock mounts several shock-mount designs have been studied. A few selected samples of the data (on five of these mounts, called mounts O, A, B, C, and D) obtained using the hammer-drop tests and the lightweight shock machine apparatus are presented here purely for the purpose of illustration.

Fig. 5 illustrates dynamic and static force-deflection curves for mount O obtained by the hammer-drop tests and on a Baldwin-Southwark Universal tester, respectively. The mount was caused to deflect at a rate of 0.15 ipm for the static tests. The time for the completion of the dynamic cycle was in the order of 0.02 sec. The straight line represents the stiffness of the mount as determined from the value of

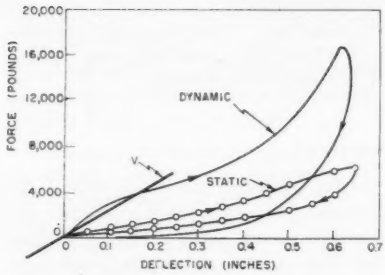


Figure 5. Characteristics of mount O. Dynamic deflections were obtained from the hammer-drop test. Slope of *V* was determined by resonance-vibration tests.

the resonant frequency when the mount was supporting a known mass. This frequency was about 35 cycles per sec (cps) and the amplitude of vibration was about 0.1 in. The rubber in this mount was especially compounded for high energy absorption.

Fig. 6 shows the averaged spectra for the four shock mounts subjected to 3-ft hammer drops on the lightweight shock machine. The deflections were in the axial direction. The shock spectrum labeled "rigid" is the averaged spectrum resulting when the "shock mount" is a block of steel. It shows the values of equivalent static acceleration to which equipment is subjected when attached rigidly to the shock machine. This figure shows that different mounts provided different degrees of shock protection. It should be noted that the spectrum from mount A crosses the spectrum for the rigid mounting at approximately 350 cps. This means that any system having a natural frequency between 50 and 350 cps would be subjected to higher stresses when mounted on A than when no mount is used.

It should be pointed out that all of the mounts except D are of conven-

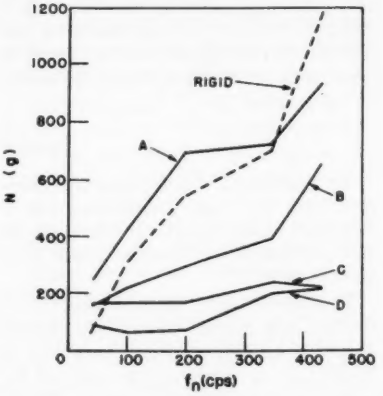


Figure 6. Comparison of averaged shock spectra for an equipment rigidly supported, or supported by shock mounts A, B, C, or D. Deflections of mounts were axial.

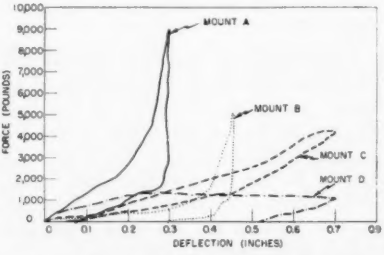


Figure 7. Axial load-deflection curves for shocks delivered by lightweight shock machine.

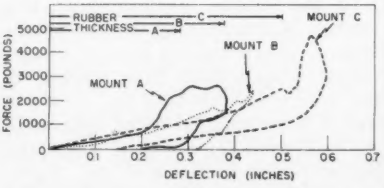


Figure 8. Radial load-deflection curves for shocks delivered by medium-weight shock machine.

tional types. Mount D is a metal strap which yields under the shock load; the energy is dissipated in the plastic deformation of the metal and is not available for subsequent vibration of the load. After the shock, the load does not necessarily return to its initial position—a characteristic regarded as unacceptable. This mount has been included in these studies because it produces the lowest shock spectrum of any mount tested.

The axial shock - load - deflection curves for mounts A, B, C, and D are shown in Fig. 6. These curves are useful in interpreting the shock spectra of Fig. 7. Mount D produced the lowest shock spectrum; the peak force was the smallest; the energy dissipated in the cycle (the area of the loop) was the greatest. Mount C produced a low shock spectrum because it remained practically linear during the loading cycle. Mounts A and B were extremely nonlinear and exerted high peak force on the load. These load-deflection curves show that both mounts A and B were overloaded. The load on mount A was 25 lb (the manufacturer's rating) but if this load had been, say, 10 lb, the load-deflection curve probably would have remained more linear.

Mount B, rated for a load of 50 lb, was tested with a load of 25 lb, which was too great for the mount. Mount C, which had a 35-lb load, was capable of absorbing the required amount of energy without "bottoming." Relatively low forces were applied to the load and a low shock spectrum resulted. The shock spectrum of mount A attains the highest magnitude of those shown, and the peak force transmitted by the mount is the greatest, reaching a value of 9000 lb. The maximum stiffness attained by mount A was about 120,000 lb per in.

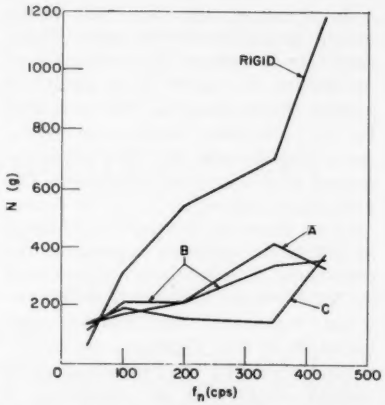


Figure 9. Averaged shock spectra for radial deflections of mounts A, B, and C.

The load-deflection curves for the mounts deflected in the radial direction are shown in Fig. 8.

The averaged shock spectra of mounts A and B for radial deflections, Fig. 9, are lower than those for the same mounts in the axial direction. This is to be expected after studying the load-deflection curves.

SUMMARY AND CONCLUSIONS

It is assumed that the probability of damage to an element of an equipment that is subjected to a shock motion increases in some manner with the maximum force experienced by that element. An idealized type of equipment was constructed which consisted of a rigid mass on which were attached a large number of single-degree-of-freedom elements. Each of these elements had a different natural frequency and together they covered the frequency spectrum of interest. This equipment was attached to one side of

a shock mount; the other side of the mount was subjected to some "standard" shock motion. The evaluation of the mount, in regard to its ability to protect the equipment, was measured by the maximum forces or accelerations experienced by the elemental masses attached to the equipment. The evaluation, therefore, can be given only in terms of frequency. A curve in which the ordinate represents the maximum accelerations experienced by the elements and in which the abscissa represents the natural frequencies of the elements, when the mount is excited by a standard shock motion, constitutes an evaluation of the protection provided by the mount. The curve frequently is called the shock spectrum and the maximum accelerations are commonly called "equivalent static accelerations." A comparison of the qualities of different mounts is made by comparing the magnitude of the shock-spectra curves obtained with different mounts.

In addition, measurements of the force transmitted by a mount as a function of deflection across the mount is determined for various standard shock motions. This defines important physical properties of the mount, such as stiffness, energy storage, and energy loss, under dynamic conditions correspondnig to use of the mount.

ACKNOWLEDGMENT

The authors wish to express their appreciation to the staff of the Rubber Laboratory of the Mare Island Naval Shipyard for their assistance relating to the hammer-drop tests. They also acknowledge their indebtedness to Messrs. John L. Bachman, Robert C. Cowan, and Arthur F. Dick, who performed many of the remaining experiments.

A SYSTEM OF RIGHT TRIANGLES

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was born at St. Louis, Mo. He was educated in the public schools of St. Louis and Webster Groves, Mo. Graduated from Northwestern University, Evanston, Ill. B.S. in Business Administration degree, with a major in advertising. His experience includes selling, exporting, and publicity and advertising. He is now engaged in free lance writing, publicity and advertising. He is a member of Northwestern University Alumni Association; and of Alpha Kappa Psi, a professional business fraternity, Northwestern University, Evanston.

The right triangle will be viewed here from a new perspective. At first, the reader may find this approach to the right triangle somewhat strange, as with anything comprising an initial experience. With familiarity, however, that feeling will soon vanish.

There will be a number of new departures in this view of right triangles. For one, we will propose a complete and integrated system of right triangles. For another, we will set forth new and original mathematical equations for the construction of right triangles. For others, there will be new concepts, laws, and equations concerning right triangles.

All right triangles, in this unified system, are divided into two types. One type is a fundamental right angle triangle, sometimes referred to here as a F.R.A.T. or a fundamental. The other type is a derivative right angle triangle, sometimes here called a D.R.A.T. or a derivative. The distinction between the two types will be brought out later.

In this system, every derivative right angle triangle is conceived as derived from some fundamental right angle triangle. It is largely for this reason that the latter has been so named, inasmuch as a fundamental right angle triangle can be visualized as a basic building block much as the atom is currently regarded.

This system of right triangles can be pictured in a two-dimensional perspective along the horizontal and vertical planes. Stretching along the horizontal axis will be the fundamentals, serving as the base of the structure. Extending along the vertical axis will be the derivatives. Theoretically, both the horizontal and vertical axes can be protracted to infinity. Since the number of fundamentals is theoretically infinite, there can be an infinite number of vertical columns rising above each fundamental. Each vertical column can also extend to infinity because the number of derivatives that can be derived from any fundamental is also infinite.

Our approach to the right triangle will be wholly in terms of its actual dimensions. When a F.R.A.T. or D.R.A.T. is mentioned, therefore, only the dimensions thereof are signified.

Before proceeding further, let us define the symbols to be used in discussing right triangles and the equations therefor.

The symbols X, Y and Z will be used to denote the three dimensions of any right triangle, whether F.R.A.T. or D.R.A.T. The symbol X will designate one side, Y will denote the other side, and Z will indicate the hypotenuse. For reasons that will become clear later, these symbols cannot be defined more specifically without setting up certain conditions—and this will be done subsequently.

The symbol M will be used to connote the concept of the multiple.

The symbols x, y and z, used only in the last section involving Diagram 1, refer to the angles of a right triangle. Those symbols have the following meanings:

x=the angle opposite side X
y=the angle opposite side Y
z=the angle opposite hypotenuse Z.

Let us now turn to the subject of the fundamental right angle triangle, in discussing which our main focal point will be the side bearing the symbol X. Indeed, we shall later determine the dimensions of both Y and Z from X by the use of formulas.

FUNDAMENTAL RIGHT ANGLE TRIANGLES

A fundamental right angle triangle is one in which the square of one of the sides equals the sum of the other side plus the hypotenuse. In any fundamental right angle triangle, therefore, the hypotenuse will exceed one of the sides by exactly 1.

A fundamental right angle triangle cannot be formed when the numeral 1 or any smaller number equals one of the two sides. To form a fundamental right angle triangle, one of the two sides must be greater than 1—even if only by .1, .01, etc. Any statement henceforth made automatically excludes, therefore, the numeral 1 or any smaller number from being set equal to one of the sides to form a fundamental right angle triangle.

Any real number that is positive can form one of the two sides of a fundamental right angle triangle, except as excluded by the previous statement. The real number forming one of the sides of a F.R.A.T. may be an integer, a fraction or a decimal—so long as it is greater than 1. When such a real number is set equal to side X, a fundamental right angle triangle can be constructed by formulas.

In any fundamental right angle triangle, when side X equals 2.42 or any larger number, then side X will be smaller than side Y. When, however, side X equals 2.41 or any smaller number, then side X will be larger than side Y.

To state the above in its most accurate terms, we should say that when side X is greater than $1+\sqrt{2}$, then side X will be smaller than side Y. And when side X is less than $1+\sqrt{2}$, then side X will be larger than side Y. The number $1+\sqrt{2}$ is an irrational number, not a real number, and is approximately equivalent to 2.4142.

When side X is equal to $1+\sqrt{2}$, then side Y will also be equal to $1+\sqrt{2}$, and hypotenuse Z will equal $(1+\sqrt{2})+1$. Therefore, when side X equals this irrational number, an isoceles right triangle is formed.

When side X equals 2.41 and 2.42 respectively, fundamental right angle

triangles with the following dimensions are formed:

Side X	*Side Y*	*Hypotenuse Z*
2.41	2.40405	3.40405
2.42	2.4282	3.4282

In view of accuracy to the hundred-thousandth in Y and Z, it seems unnecessary from a practical standpoint to have X equal any number having a decimal finer than hundredths.

To clarify the foregoing situation, therefore, fundamental right angle triangles have been divided into two classes: a COMFRAT class and an INFRAT class. These names were chosen to describe the facts. In the COMFRAT class, by far the most common, side X will be the short side. In the INFRAT class, examples of which are met infrequently, side X will be the long side.

In the above illustration, an INFRAT is formed when side X equals 2.41, and a COMFRAT is constructed when side X equals 2.42. It becomes obvious, therefore, that only numbers between 1.01 and 2.41, when set equal to side X, can form INFRATS. That range is indeed very narrow and includes only one integer, the numeral 2. By and large, therefore, side X will be the short side in an overwhelming number of instances.

Some typical fundamental right angle triangles in which side X equals a whole number are: 3-4-5, 5-12-13, 7-24-25, 9-40-41, 4-7½-8½ and 6-17½-18½. Each of these is a COMFRAT inasmuch as the short side, squared, equals the sum of the long side plus the hypotenuse.

In summary, the salient properties and characteristics of a fundamental right angle triangle are:

1. The square of one of the sides of a F.R.A.T. equals the sum of the other side plus the hypotenuse.
2. The hypotenuse of a F.R.A.T. exceeds one of its sides by exactly 1.
3. The sum of the dimensions of any F.R.A.T. is equal to $X+X^2$, where X equals one of its sides.
4. Any number from 1 to infinity can be set equal to side X, thereby enabling a F.R.A.T. to be formed.
5. Any multiple of a F.R.A.T. is itself a right triangle.
6. In a COMFRAT the short side, squared, equals the sum of the long side plus the hypotenuse and the latter exceeds the long side by exactly 1.
7. In an INFRAT the long side, squared, equals the sum of the short side plus the hypotenuse and the latter exceeds the short side by exactly 1.
8. When X, the side squared, is equal to 2.42 or any larger number, a COMFRAT will result with side X being the short side.
9. When X, the side squared, is equal to 2.41 or any smaller number greater than 1, an INFRAT will result with side X being the long side.

Let us now look at the formulas for fundamental right angle triangles.

FORMULAS

1. In any F.R.A.T., when one side or X equals any numeral greater than 1, then:

(a) $Y=\frac{X^2-1}{2}$

(b) $Z=\frac{X^2+1}{2}$.

2. In any COMFRAT, when one side or X equals 2.42 or any larger number, then that will be the short side, and

(a) $Y=\frac{X^2-1}{2}$

(b) $Z=\frac{X^2+1}{2}$.

3. In any INFRAT, when one side or X equals 2.41 or any smaller number, then that will be the long side, and

(a) $$Y = \frac{X^2 - 1}{2}$$

(b) $$Z = \frac{X^2 + 1}{2}.$$

HOW TO CONSTRUCT F.R.A.T.'S QUICKLY BY THE F.R.A.T. LAW OF PROGRESSION

With one exception, a series of F.R.A.T.'s is shown in Table 1. To the left are those formed when consecutive odd numbers represent X; to the right are those formed when consecutive even numbers represent X.

It will be noted that the figures in column X increase successively by two, whereas the figures in columns Y and Z increase by a difference equal to X plus the X immediately following. For example, the increase in Y and Z from 11-60-61 to 13-84-85 equals 24, or 11 plus 13.

TABLE 1

One Side X	Other Side Y	Hypotenuse Z	One Side X	Other Side Y	Hypotenuse Z
1	0	1	2	1½	2½
3	4	5	4	7½	8½
5	12	13	6	17½	18½
7	24	25	8	31½	32½
9	40	41	10	49½	50½
11	60	61	12	71½	72½
13	84	85	14	97½	98½
15	112	113	16	127½	128½
17	144	145	18	161½	162½
19	180	181	20	199½	200½

Table 1 was constructed partly by formula and partly by the law of progression for F.R.A.T.'s.

LAW OF PROGRESSION

By the F.R.A.T. law of progression, when X is represented by a consecutive sequence of whole numbers, X increases constantly by 1, while both Y and Z increase by a difference equal to ½ less than the figure representing X, as shown in Table 2.

TABLE 2

One Side X	Other Side Y	Hypotenuse Z	Increase in both Y and Z
1	0	1	—
2	1½	2½	1½
3	4	5	2½
4	7½	8½	3½
5			4½
6			5½

The law of progression, therefore, makes possible the construction of an indefinite table of F.R.A.T.'s without resorting to the formulas, when X's are integers.

Some variation of this law will be found to exist when X is represented by a consecutive sequence of fractional or decimal numbers.

PERIMETER AND AREA FORMULAS FOR ANY F.R.A.T.

In any F.R.A.T. the perimeter and area can also be determined by equations. As before, when any number is set equal to X the formulas can be used. The meaning of the symbol is: P equals perimeter, A equals area, and X again equals any number not previously excluded.

$$P = X + X^2$$

$$A = \frac{X^3 - X}{4}$$

DERIVATIVE RIGHT ANGLE TRIANGLES

A derivative right angle triangle is one which is derived from a F.R.A.T. as some multiple thereof. Stated inversely, any multiple of a F.R.A.T. is itself a right triangle, i.e., a derivative right angle triangle.

Any fundamental right angle triangle can serve as a base to form a chain or family of right triangles, that is, derivative right angle triangles. Such a chain of right triangles, all derived from the same F.R.A.T., will have proportions identical with each other and with those of the F.R.A.T. from which they are derived. That is to say, all right triangles in such a chain will be similar. In such a chain each D.R.A.T., derived as a variant multiple of the same F.R.A.T., will have dimensions different from those of all the others. This is to say, no two in the chain will be congruent, or identically equal. Within the same chain, therefore, right triangles are similar but not congruent.

When two separate chains of derivative right angle triangles are formed from two different fundamentals, those in the one chain are neither similar to, nor congruent with, those in the other chain. Similarity ceases outside of the chain and congruency does not exist even within the chain. The previous statement as regards congruency presupposes that no two right triangles are duplicate multiples of the same F.R.A.T.

THE CONCEPT OF THE MULTIPLE

A derivative right angle triangle can be formed when the dimensions of a F.R.A.T. are either magnified or contracted. To magnify, we multiply the dimensions of a F.R.A.T.; to contract, we divide its dimensions.

When we magnify the dimensions of a F.R.A.T., we multiply its dimensions by some number greater than 1, and when we contract its dimensions we take some fractional part of its original dimensions. Hence, when we magnify the dimensions of a F.R.A.T., the multiple is greater than unity (1); when we contract its dimensions, the multiple is less than unity (1). The term "multiple" is thus used here to reflect the process whereby the dimensions of a F.R.A.T. are both magnified and con-

tracted to form a derivative right angle triangle.

The 5-12-13 F.R.A.T., for example, cna be magnified into a right triangle of 20-48-52 by multiplying by 4, or it can be contracted into a 2½-6-6½ right triangle by dividing by 2, i.e., multiplying by ½. In each instance, the multiple is reflected in the dimensions of the derivative right triangle as the difference between the hypotenuse and one of the sides. In the above illustrations, 52 minus 48 is 4, the multiple; and 6½ minus 6 is ½, the other multiple.

With any F.R.A.T. before us, we may thus magnify or contract it to any desired dimensions by our selection of the multiple. Inversely, with any D.R.A.T. before us, we can determine exactly what multiple formed it from a F.R.A.T. For example, with a D.R.A.T. of 50-120-130, the fact that it is the 10th multiple of some F.R.A.T. is reflected in the difference between 130 and 120. After dividing 50-120-130 by 10, we find that this right triangle is the 10th multiple of the 5-12-13 F.R.A.T. This principle is the same with all fundamental and derivative right angle triangles.

In every derivative right angle triangle formed from a COMFRAT, the multiple is reflected in the difference between the hypotenuse and the long side. In any D.R.A.T. formed from an INFRAT, the multiple is the difference between the hypotenuse and the short side. As we have seen the number of possible INFRATS is greatly limited and circumscribed, thereby also limiting the number of derivatives that can be formed therefrom. We may therefore say that, almost always, a derivative right angle triangle reflects its multiple as the difference between the hypotenuse and the long side.

The concept of the multiple performs an important function in our system, as will shortly be seen in the equations for any right triangle and in the various ways in which those equations can be used. For this concept a symbol was needed and, as previously indicated, the symbol M designates the multiple.

HOW TO DETERMINE A RIGHT ANGLE TRIANGLE BY THE F.R.A.T. SYSTEM

Any three dimensions reputed to represent a right angle triangle can be quickly tested in a simple manner. For example, assume we are given the dimensions of 45-108-117 and asked whether they form a right triangle. The answer by steps is as follows:

Step #1

Find the multiple, which is here the difference between the hypotenuse and long side. Multiple equals 117 minus 108 or 9.

Step #2

Divide all dimensions by the multiple.

$$9 \mid \underline{45 - 108 - 117}$$
$$\quad\;\; 5 - 12 - 13$$

Step #3

If result of Step #2 is a F.R.A.T., the answer is yes; if not, the answer is no.

5, squared equals 12 plus 13, and hypotenuse exceeds long side by exactly 1. Therefore, since 5—12—13 is a F.R.A.T., the answer is yes. The 45—108—117 right triangle is a D.R.A.T., being the ninth multiple of the 5—12—13 F.R.A.T.

HOW TO DETERMINE TWO UNKNOWN DIMENSIONS OF ANY RIGHT TRIANGLE FROM TWO FACTS

With the F.R.A.T. method, one can determine two unknown dimensions of any right triangle from two facts concerning it. The facts needed would be 1) one side (usually the short side), and 2) the multiple.

For any right triangle, the following formulas hold true. In the formulas, X equals one of the sides, Y equals the other side, Z equals the hypotenuse, and M equals the multiple or hypotenuse minus side Y. (When, as in the example below, we are given the dimensions of one side, the multiple is the difference between the two unknown dimensions, Y and Z.)

FORMULAS FOR ANY RIGHT TRIANGLE

$$(1) \qquad \frac{X^2}{M} = Y + Z$$

$$(2) \qquad \frac{X^2 - M^2}{2M} = Y$$

$$(3) \qquad \frac{X^2 + M^2}{2M} = Z$$

Let's assume a right triangle with dimensions, unknown to us, of 120-225-255, or X, Y and Z respectively. If given two facts about that right triangle—one side and the multiple—it is possible to name the other two dimensions. These two facts would be 1) 120 (side X) and 2) 30 (multiple). Then, by formula

$$\frac{X^2}{M} = \frac{120^2}{30} = Y + Z.$$

Since we know that Z equals $Y+M$ ($Y+30$ here), we therefore know that $Y+Z$ equals $2Y+M$ ($2Y+30$ here), and we may either solve two linear equations or use the other two formulas to obtain Y and Z.

The 120-225-255 right triangle is the 30th multiple of the 4-7½-8½ F.R.A.T.

HOW TO MAKE YOUR CHOICE OF A RIGHT TRIANGLE

Let's imagine that you want to custom fit at least one side of a right triangle to fit your needs, and, being selective, you would like a choice among several right triangles regarding the other side and hypotenuse.

Let's assume that you want one side of 40. A simple way to meet the problem is with a technique that is shortened by telescoping the F.R.A.T. and D.R.A.T. systems.

A little reflection will tell you that 40 is an integral multiple of various numbers. Since 40 is divisible by 2, 2½, 4, 5, 8, 10 and 20, it is the 20th, 16th, 10th, 8th, 5th, 4th, and 2nd integral multiple of those numbers, respectively. This means that there are F.R.A.T.'s with side X equal to 2, 2½, 4, 5, 8, 10 and 20 which can be magnified into D.R.A.T.'s with side X equal to 40. Therefore, you already know that, aside from a F.R.A.T., there are at least seven (7) possible right triangles (D.R.A.T.'s) from which you can select. What are their dimensions?

TABLE 3

M or Multiple	X^2/M	$Y+Z$	$Y+Z$ equals $2Y+M$ or	$2Y$	Y	Z
20	1600/20	80	2Y+20	60	30	50
16	1600/16	100	2Y+16	84	42	58
10	1600/10	160	2Y+10	150	75	85
8	1600/8	200	2Y+8	192	96	104
5	1600/5	320	2Y+5	315	157½	162½
4	1600/4	400	2Y+4	396	198	202
2	1600/2	800	2Y+2	798	399	401

Recalling the formula stating $X^2/M = Y+Z$, and remembering that Z equals $Y+M$, you can set up a table similar to Table 3 and quickly arrive at each of the other two dimensions (Y and Z), knowing that one side in each right triangle will be 40. You are then free to make your choice.

In only one case is Y (30) smaller than X (40). The reason for that is the 40-30-50 right triangle is the 20th multiple of the 2-1½-2½ F.R.A.T., which is an INFRAT inasmuch as the long side, squared equals short side plus hypotenuse. All the other right triangles are multiples of some COMFRAT.

We have here constructed a number of right triangles in which side X has been held constant at 40 while we have varied Y and Z at will by our selection of the value for M.

HOW TO USE THE RIGHT TRIANGLE EQUATIONS

In the foregoing formulas 1, 2 and 3 we need only know two facts: the length of side X and the multiple, which is Z minus Y. In point of fact, we need not *know* even those two facts to use the formulas. Instead of known facts, we may use our imagination to reflect unknown facts inasmuch as there are many right triangles whose dimensions are not now known. Indeed, right triangles—like the stars—exist by the millions or billions, although the existence of but relatively few is known to us.

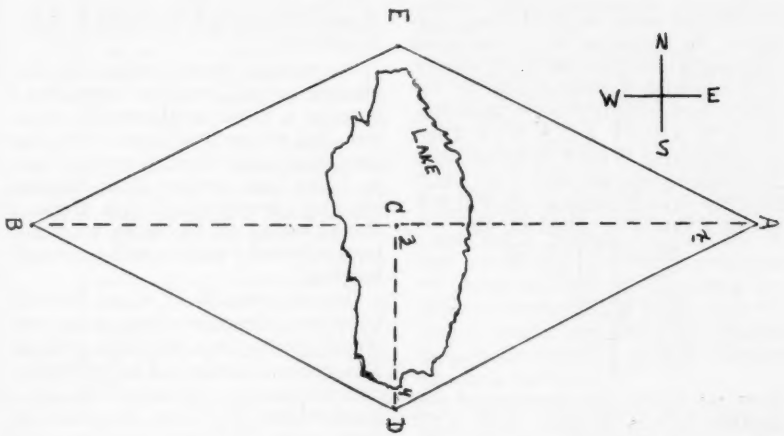
We can substitute in the formulas any desired value for X and then imagine some value equal to M, thereby achieving as a final result a right triangle. The resultant right triangle will depend on our values for X and M. It is predictable, however, that the resultant right triangle will be some multiple of that F.R.A.T. having as its side X the value of X/M. For example, assume that in Formula No. 1, we substitute 15 for X and imagine 3 for the value of M. Having done so, we can predict that the right triangle resulting therefrom will be the third multiple of that F.R.A.T. having 5 as the dimension of its side X, since 15/3 (X/M) equals 5. Only one F.R.A.T. can have its side X equal to 5 and that is the 5-12-13 F.R.A.T. With this knowledge, we can instantly conclude that the solution of the equation X^2/M equals $Y+Z$ will here result in a D.R.A.T. with dimensions of 15-36-39.

When using our imagination to substitute values for X and M in Formulas 1, 2 and 3, we must do so realistically. Since, in any right triangle the value of M cannot equal or exceed the value of X, we cannot reasonably or realistically imagine as our value for M any numeral that is equal to or greater than the value substituted for X. That is to say, in any right triangle each of the sides must be larger than the difference between the hypotenuse and the other side, viz., X must be greater than Z minus Y, and Y must be greater than Z minus X.

We can illustrate the above point by letting the decimal numeral 1.2 equal X, thereby forming one of the very smallest of possible fundamental right angle triangles. The resultant F.R.A.T. will thus have dimensions of 1.20-.22-1.22 for X, Y and Z respectively. We can see that the previous statement holds true, since 1.20 (X) is greater than 1.00 (Z minus Y) and .22 (Y) is greater than .02 (Z minus X).

HOW TO CONSTRUCT ROADS AND RAILROADS ECONOMICALLY

The cost of building a road or railroad consists of labor and materials and is usually proportionate to its length. Constructing the shortest pos-



C-D = X
A-C = Y
A-D = Z

Figure 1

sible and most feasible route, therefore, will effect economies in construction expense—and may also result in greater efficiency of operation as well.

The railroad built across the Great Salt Lake may, perhaps, be the shortest possible route. It is questionable, however, whether that route is the most feasible, for it definitely is not efficient in operation inasmuch as trains must slow to about 5 miles-per-hour when crossing it. In a day when trains average about 60 miles-per-hour, the annual loss in time and money chargeable to inefficiency represents a considerable sum. Moreover, the railroad across that lake is a potential bottleneck because, if damaged, it would take much longer to repair than a railroad constructed across land terrain. Furthermore, such a railroad would be an ideal target for any air force since bombs that were merely near hits would weaken the structure so that the weight of the next train that crossed it might well crumple it.

Let us imagine that we plan to construct a railroad, in a east-west direction, between points A and B as shown in Diagram 1. Let us assume that about midway between points A and B is located a natural barrier, say, a lake 30 miles long from north to south and 20 miles wide from east to west. The shortest possible distance from A to B is directly across this lake and is 225 miles, plus or minus 1 mile. To construct 20 miles of railroad across the lake would involve great construction costs—perhaps as much as from three to ten times the normal cost-per-mile. Naturally, we wish to avoid this great expense if at all possible.

Can we blueprint in advance a short, feasible and efficient route for the railroad and yet avoid that 20-mile stretch across the lake? Yes, we can.

Let's assume that point C, somewhere in the lake, is equi-distant from A and B. We know, or determine, that the distance C-D equals 15 miles, with point C being on a straight line from A to B. At point C in the lake a buoy with a large balloon streaming from it would be left as a marker for future use. Knowing that distance C-D

equalled 15 miles, we would then construct, by formula, a fundamental right angle triangle in which dimension A-D exceeded dimension A-C by exactly 1. In this right triangle, therefore, C-D= X, A-C=Y, and A-D=Z. Then, by substituting 15 for X in the equation hereinafter given and solved, we would have the three dimensions of a F.R.A.T. in which side X equalled 15, viz.,

$$X^2=2Y+1=Y+Z$$
$$15^2=2Y+1$$
$$225 =2Y+1$$
$$224 =2Y$$
$$112 = Y$$
$$113 = Z$$

Thus, we would have determined that distance C-D is 15 miles, A-C is 112 miles, and A-D is 113 miles. Likewise, distance C-B is 112 miles and D-B is 113 miles. We now know that, for the distance of 224 miles in a straight line from A to B, the shortest possible distance around the lake barrier is equal to twice 113 miles or 226 miles.

Having determined the triangle dimensions of 15-112-113 miles, we would then construct this right triangle very accurately in miniature—say in inches or feet. With a theodolite (transit), we would then measure angle x at point A and angle y at point D, as indicated in Diagram 1. We would then know all three dimensions and all three angles of the right triangle embracing the proposed route for the railroad.

We would have a choice, however, of building to the north or south of the lake to connect points A and B. To determine our choice, we would reconnoiter in each direction, scientifically and efficiently.

We would select either an airplane or a helicopter for an aerial survey. Or, we might choose both of them, using the former for a first and rapid survey and the latter for a later and slower survey to gather important construction data. In either event, we would set out to fly from B to D, D to A, and then from A to E, and E to B—scientifically.

We already know angle x at our starting point B, and we know that it runs in a north-northeasterly direction. But we want to know more than that prior to our flight. We would want to know just exactly what compass heading corresponded with angle x. Before taking off, therefore, we would have correlated angle x with a compass heading.

Ground points B, D, A and E would have been clearly marked so as to be plainly visible from our flight altitude. The markers used could be brilliantly-colored weather balloons tethered a hundred feet or so from the ground at those points. Or, these markers might be some device such as a slow-burning smoke bomb that could be either placed or dropped from the air at the desired locations. Point E, for example, might not have been reached or explored on foot. If so, how then could it be marked for our survey flights?

One way of marking point E would be to have a control flight made from point D to point C in the lake. The control flight would be made at constant air-speed and the elapsed time of flight carefully checked. The control flight could then be duplicated, on a time basis, from point C to point E, where a marker such as a smoke bomb could be dropped.

These markers, being observable from the air, would serve two purposes during our survey flight. They would pinpoint for us the desired ground points, indicating the need to alter our course of flight. Also, these markers would serve as a check on our compass heading after each change of course, thereby aiding the pilot to navigate with the purpose of holding his line of flight directly above the proposed railroad route.

Thus set up and equipped, we could make survey flights directly over the exact route contemplated for the rail-

road. Knowing that the distance between each of the four ground markers was 113 miles in a east-west direction, we could conclude that each of the alternative routes could be quickly surveyed during an hour's flight by using a reasonably fast plane. Or, using a helicopter for a complete aerial survey, we could allow 4 hours to map each of the alternative routes. The previous statement is predicated on the assumption that the helicopter pilot would be given instructions to fly at a constant air-speed equivalent to 60 land miles per hour. Flying at that speed and aided by our watches, we could then correlate our air position with the exact land position directly below us on the proposed railroad route. Each minute of elapsed flight time would, therefore, indicate the exact number of land miles from starting point B and from each of the other points after successive changes in course.

For the sake of simplicity, we have here assumed that points A and B were equally distant from point C. However, that assumption is not necessary and, in fact, will seldom be the actual case. Even when the respective distances to the barrier are unequal, the principal cited is the same. Some fundamental right angle triangle can always be found to fit any given set of conditions or distances. In most situations, two different fundamental right angle triangles would very likely be required to give the shortest possible distance around a barrier.

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SEMICONDUCTORS AND THE TRANSISTOR

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INTRODUCTION

The transistor has made many scientists in unrelated fields aware of the importance of a class of materials known for many years to the physicist. These substances, known as semiconductors, have electrical resistance mid-way between insulators and metals, and this fact alone provides the only workable definition. Metals have their electrical resistivity expressed in microhm-cms., that is, millionths of an ohm-cm., while good insulators have their resistivity expressed in millions of megohm-cms. This range of 10^{-6} to 10^{+12}, or 18 orders of magnitude, encompasses a wide variety of possibilities, of which the range of 10^{-3} to 10^{+6} ohm-cms. is perhaps of greatest practical interest for semiconduction.

What is the predominant factor which makes the semiconductor of such interest today? Modern technology is based on control of much by little, control of many by few; in electronics, this is called "amplification" or the ability of a small input to control a large output. The electron tube was, perhaps, the first major advance and the transistor appears to be the second. In the tube, one controls free electric charge carriers in a vacuum or rarefied gas, in the transistor these electric charges stay inside a solid materials. But again, why does one need the semiconductor? This is quite simple, really, since, in a metal, the free charges which carry electric current are much too plentiful to be readily controlled by a low power. In an insulator, the free charges may not normally be present and so, again, control is difficult. Only in the semiconductor is the number of free charges neither too great, nor too few, and one is able to build an "amplifier." Interestingly enough, the solid-state photodevice utilizes approximately the same resistance range, so that photoconduction and semiconduction are but two aspects of an essentially similar class of materials.

In the present discussion, the main emphasis will be on the transistor, with relatively little review of the older and simpler devices such as the diode rectifier or the photodevice. For background, a short historical survey will be given outlining some of the milestones on the way. Then the basic principles of semiconduction will be discussed. The techniques, with which the two currently important elements, germanium and silicon are applied to the diode and the transistor, will also be reviewed. The discussion closes with a description of present commercial and laboratory developments and some views into the future of this new art.

HISTORY

Historical background can be divided into two aspects, the first concerned with actual experimental devices, and the second about man's understanding of these devices.

The beginning of the modern semiconductor device came in the early 1870's when Ferdinand Braun (1)* observed electrical nonlinearity, that is, rectifying action, in certain metallic sulphides such as galena and pyrites. At about the same time selenium became of interest and, by 1883, Fritts devised a selenium rectifier which already had modern features (2). The advent of wireless telegraphy, or radio as it is now called, led to an extensive exploration of the sulphides and many other natural minerals as radio detectors (3). The first recognition that the element, silicon, could be used was due to Pickard (4), about 1906. As early as 1911, Merritt (5) used silicon as a detector of radio waves at the (then) phenomenally short wavelength of 15 cm. (2000 megacycles per second). Today, the silicon crystal detector still reigns supreme for use at the highest frequencies, called microwaves, and has been used up to 60,000 megacycles and higher.

The 1920's saw a very wide use of the crystal detector in early broadcast reception, and it is rare to find anyone over 40 today who does not remember the experience of searching for a sensitive spot with a "cat-whisker" on a piece of galena. About 1926, another element, germanium, was added to the list of useful rectifying materials (6). This same decade saw not only the announcement of the large-area copper oxide rectifier (7) but the first real appreciation of the role the semiconductor could play in the more significant area of amplification, with the filing of some patents by Lilienfeld (8). The 1934 patent application of O. Heil (9), however, shows the first glimpse of a semiconductor triode amplifier device which can be said to be modern in concept. Unfortunately, the techniques available in the 1920's and 1930's were still inadequate to support the ideas which were advanced. The only published record of any experimental results on a solid-state amplifier was the Hilsch and Pohl potassium-bromide crystal amplifier of 1938; this used ionic phenomena and was too slow in action for practical application (10).

Coming now to the World War II period, the silicon crystal, in improved form, made possible microwave radar, and interest in the relatively unexplored semiconductor element, germanium, was revived. Simultaneously in Germany and in the U. S. germanium was investigated and its importance became established by Benzer's dis-

*The numbers in parentheses refer to the references at the end of this article.

covery (11) of the high-back-voltage point-contact rectifier, which overcame the low-voltage limitations of earlier crystals. This war period was significant also in the abandonment of natural minerals for detector devices, substituting controlled synthesis and purification of the two most suited elements, germanium and silicon (12).

A really major step was taken in 1948 when Brattain and Bardeen announced that, by addition of a second point contact to a germanium of silicon rectifier, an amplifying triode point-contact transistor could be produced (13). At about the same time, Shockley applied for a patent (14) on a conceptually more elegant device which we call today the junction transistor, and which seems more likely to be the basis of future devices.

This brings us to the present era, as far as devices are concerned, and we are ready to examine the second aspect of history in this field. Only after more than 50 years of work with semiconductors did man's understanding of their electrical behavior begin to take shape. Perhaps the start came in 1928 when Sommerfeld (15) showed that the new wave mechanics could be used to explain conduction in metals. Shortly thereafter, another German physicist, Gudden (16), recognized that the conduction in semiconductors was almost entirely due to the impurities in them, rather than to their intrinsic properties. This, of course, was highly significant but became much more so in 1931 with Wilson's work establishing the modern "hole" and electron concepts of semi-conduction (17). Less than ten years later, the first theoretical pictures of rectification were published, for the semiconductor-to-semiconductor junction (now known as p-n junction) by Davydov (18), and for the metal-to-semiconductor

IA	IIA	IIIB	IVB	VB	VIB	VIIB	VIII			IB	IIB	IIIA	IVA	VA	VIA	VIIA	INERT GASES
																H	He
Li	Be											B	C	N	O	F	Ne
Na	Mg											Al	Si	P	S	Cl	A
K	Ca	Sc	Ti	V	Cr	Mn	Fe	Co	Ni	Cu	Zn	Ga	Ge	As	Se	Br	Kr
Rb	Sr	Y	Zr	Nb	Mo	Tc	Ru	Rh	Pd	Ag	Cd	In	Sn	Sb	Te	I	Xe
Cs	Ba	La	Ce / Hf	Ta	W	Re	Os	Ir	Pt	Au	Hg	Tl	Pb	Bi	Po	At	Rn
Fr	Ra	Ac	Th	U													

Figure 1. The shaded elements of the periodic table are semiconductors. Of great interest today are Ge, Si and many two-element combinations in and near column IVA such as GaAs, InP, CdS and Ge + Si.

junction by Schottky (19). The war period of 1939-1945 led to many advances but little publication, and it is difficult to pick out the important milestones. The monumental theoretical studies by Shockley and his colleagues (20), in the post-war period, may be said to complete the background for the rectifier and transistor, as we know it today.

It is now appropriate to discuss the present status of the semi-conductor, and how it is used to make a rectifier or transistor.

SEMICONDUCTORS

The modern "band" theory gives acceptable explanations for most observed characteristics of semiconductors. In an insulator, there is a large "forbidden" band between the energy levels occupied by the electrons of the atoms and the next possible band, so the electrons are not free to move. In metals, there is no such band, while in semiconductors the band is intermediate, that is, it ranges from 0.1 electron volt to perhaps 3 volts for different materials. In a pure semiconductor, the only conductivity is due to electrons and holes produced by thermal energy. Greater conductivity can be provided by addition of impurities which may be of either electron-giving, that is, donor type, such as P, As, Sb, Bi, etc., or electron-collecting, that is, acceptor type, such as Ga, In, B, etc. For transistors, no use is made of the thermally-produced conduction and, indeed, it is not at all desirable because it varies so much with temperature. Instead, n-type and p-type semi-conductors are used, whose conductivity is due to impurities of donor or acceptor type, respectively.

Of great interest for transistors are two properties of a semiconductor.

1. The band gap, which is a measure of how much intrinsic (undesired) conduction is present. A large band gap is desired.

2. The electron and hole mobilities, which are a measure of how fast these charge carriers move in an electric field. A high mobility is desired.

Unfortunately, nature seems to have provided a relationship between band gap and mobility in many known semiconductors so that when one goes up, the other goes down. Mankind is presently trying to overcome this by a search for new materials. The problem is made more difficult in that maximum mobility can be achieved only in nearly-perfect crystals.

Examination of the periodic table, Fig. 1, is of interest. Only a small number of elements are available as crystalline semiconductors useful for transistors. For example, in Column IV, as we go up from tin to germanium to silicon to diamond, we have higher melting points and larger band gaps but smaller mobilities. Tin is too close to a metal and diamond is too close to an insulator. This leaves us only germanium and silicon for the transistor. There are certain borderline elements such as boron, selenium, tellurium, etc., which are potentially semiconductors but these are either not readily available in crystalline form or do not have suitable band gaps.

Among the mixtures and compounds, there is an infinite variety but none has been made in pure enough, or perfect enough form, so that its ultimate properties as a semiconductor are known. For the future, certain ones are of particular interest. First, we have the Column IV combinations, silicon carbide, germanium and carbon, and germanium and silicon. Each may be expected to have intermediate properties between its constituents and

so lend greater variety to the simple valence crystals. Then we have the less homopolar crystals from Column III combined with Column V. One reads in the literature (21) about indium antimonide, gallium arsenide, indium phosphide and aluminum antimonide; perhaps one of these will provide a higher value of mobility for a given band gap. One also hears much about the Column II-Column VI combinations, cadmium sulphide, cadmium telluride, zinc selenide, etc. Finally, there are the traditional crystal detector compounds, the sulphides of lead, molybdenum, iron, etc. We must reserve judgment on all these compounds until they have achieved a higher state of development.

For the present, we are left with germanium and silicon as of greatest importance. It is of interest to compare their characteristics as shown in Table I. The effect of the larger band gap of silicon in increasing its intrinsic (pure) resistance at room temperature is clearly indicated. It is because

TABLE I.—*Properties of Germanium and Silicon*

	Band Gap, electron volts	Resistivity ohm-cm. Intrins. (Pure)	Resistivity ohm-cm. Useful (Impure)	Mobility cm^2./volt-sec. Hole	Mobility cm^2./volt-sec. Electron	Max. Temp., °C.
Ge	0.72	60	.1—10	1700	3700	100
Si	1.12	10,000	.1—10	300	1200	250

of this that silicon is useful to a much higher temperature than germanium. Unfortunately, one sacrifices something in mobility, so that silicon devices will not ordinarily have quite as good performance as germanium ones, when the latter are kept low in temperature. For this reason, it is likely that germanium will remain the most widely used substance for some time to come, though Silicon may supplant it in special applications.

MODERN TECHNIQUES

Germanium and silicon are first purified chemically, and then by a zone melting process (22). When this stage is reached, there are no conventional means for measuring the impurity content, because it is so small. Instead, an indirect measure is used, the resistivity, using a curve as shown in Fig. 2, taken from a calculation by Herkart and Kurshan (23). Ordinarily we can reach about 40 ohm centimeters, which corresponds to one part impurity for each billion atoms of germanium. This is about ten or more times purer than necessary for transistors or rectifiers, which use resistivities between 0.1 and 10 ohm centimeters.

For transistor or rectifier use, minute quantities of the appropriate *n* or *p* type impurities are added to a silicon or germanium melt and a single crystal is grown by the Czochralski technique (24). With this method, a cold seed crystal touches the melt and, as crystallization starts, the seed is slowly withdrawn (at about 1 mm. per minute) pulling with it a single-crystal bar of germanium or silicon. The perfection of the crystals of germanium grown in this way can be truly

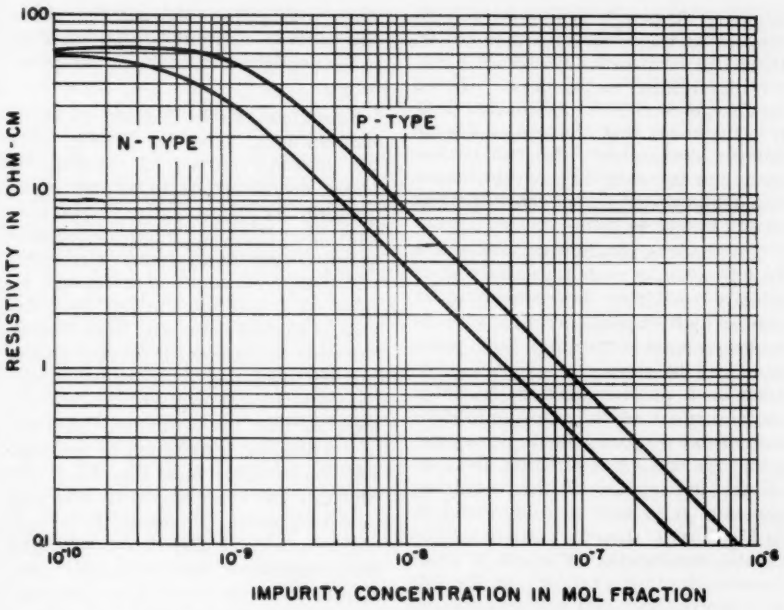


Figure 2. Theoretical resistivity of germanium at room temperature as a function of impurity content. Such curves are often used to estimate purity.

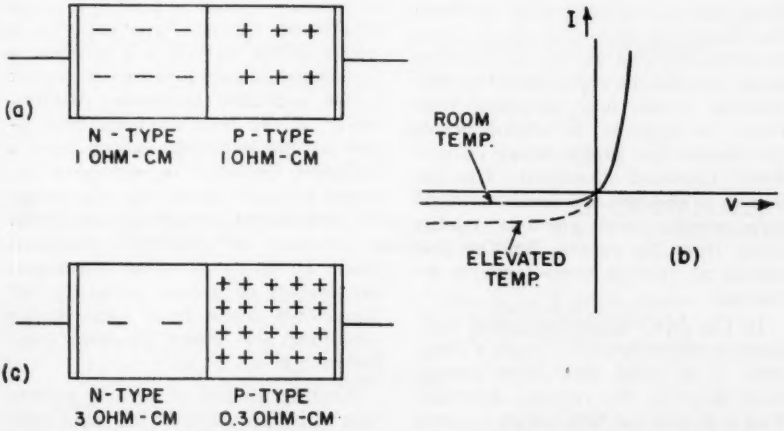


Figure 3. The rectifying action in a p-n junction; (*a*) a junction of equal resistivity parts; (*b*) a junction of unequal resistivity parts with most current carried by holes; (*c*) rectifying characteristic of either type.

amazing. Although impurities may be present of the order of a part in 10^8, these are substitutional and not a major disturbance to the crystal. Based on charge carrier recombination data, it is believed that the major crystal defects, dislocations, etc., can be less than one for each 10^{11} parent atoms, and the crystals are as nearly perfect as any known to man.

The modern rectifier or transistor is based on using such crystals in electrically rectifying junctions, that is, contacts which pass current in one direction but not in the other. Such junctions can be metal-to-semiconductor, such as a point contact or a plated metal contact, or they can be a junction of two regions in a single crystal, which we call a p-n junction. Because of the importance of the rectifying junction, it is well to understand it. In Fig 3 (a) is shown a piece of single-crystal semiconductor which is electron-conducting (n-type) to the left of a central line, and hole-conducting (p-type) to the right. If the impurities are such as to produce about one ohm-centimeters resistivity on each side, one may schematically represent the charges as shown, in nearly equal numbers. When a battery is connected so as to make the right-hand terminal positive, holes and electrons both move in opposite directions across the central line and a heavy current flows (forward direction). The reversal of the battery leads to slight hole motion and electron motion *away* from the central line, so that almost no current flows (reverse direction).

In Fig. 3(b) is the rectifying volt-ampere characteristic of such a junction. It is noted that some current does flow in the reverse direction. This is due to the thermally generated holes and electrons throughout the crystal, and these increase the temperature. For germanium, the currents are appreciable, but for silicon the reverse currents are so small as to be negligible at ordinary temperatures.

Junctions can be formed of many combinations, however. In Fig. 3(a) the forward current is carried by holes and electrons in almost equal quantity. In Fig. 3(c), the n-type portion has reduced impurity content (higher resistance, fewer electrons) while the reverse is true for the p-type. As a *rectifier*, the Fig. 3(c) device is just like the Fig. 3(a) device. But now the current is carried almost entirely by holes. In a *transitor*, one controls current carriers of only one sign, so that the so-called "emitter" of a transistor is designed to resemble Fig. 3(c) rather than Fig. 3(a), except that the ratio of resistivities may be even greater. Of course, the n-type and p-type roles may be interchanged and transistors can use either predominantly hole current, or predominantly electron current.

It is now appropriate to consider how rectifying junctions can be made in practice. If during pulling a single crystal, an opposite type impurity is added to the melt, a p-n junction is formed; a double junction, p-n-p, or n-p-n can also be made. Alternatively, if the melt contains both p- and n-type impurities which have a different tendency to segregate between solid and liquid, one can change the growth rate suddenly, and obtain a reversal of impurity. Junctions made by either of these techniques are known as *grown* junctions and transistors made from such double junctions are *grown junction* transistors. (25).

Another method of forming a junction is by diffusion in the solid state (26). An impurity is evaporated or plated on the surface of a crystal al ready containing an opposite impu-

rity. Heating over a period of time causes the surface impurity to diffuse inward, producing a junction.

A commonly used method of producing a junction however, is by a still different process called fusing or alloying (27). To illustrate the method, let us assume uniform n-type germanium, with a junction to be made by means of a p-type impurity such as indium. The steps are shown in Fig. 4(a) to 4(d). In Fig. 4(a) a small piece of indium is placed on the germanium. The temperature is raised just above the melting point of the indium as in Fig. 4(b). Germanium is slightly soluble in liquid indium, the solubility rising as the temperature goes up. By raising the temperature to about 500° C., as in Fig. 4(c), enough germanium can be dissolved to leave a depression in the surface, with a molten indium-germanium solution on top. Subsequent slow cooling causes the germanium to recrystallize on the original unmelted single-crystal base, growing a new indium-contaminated (p-type) crystal as a continuation of the original base. The p-n junction forms at the original solid-liquid interface and the last material to solidify, the indium, makes an excellent connection to the junction. The p-type germanium is so low in resistance that the majority of the current is carried by holes, so that a good emitter for a transistor can be made.

As already indicated, one can also make a rectifying contact analogous to that of the p-n junction by means of a metal-to-semiconductor contact. This is the basis of the early, and the modern, crystal detectors in which a pointed wire was placed against a crystal. Point-contact techniques, without junction formation, are used for transistors as well. However, one does not usually obtain current flow

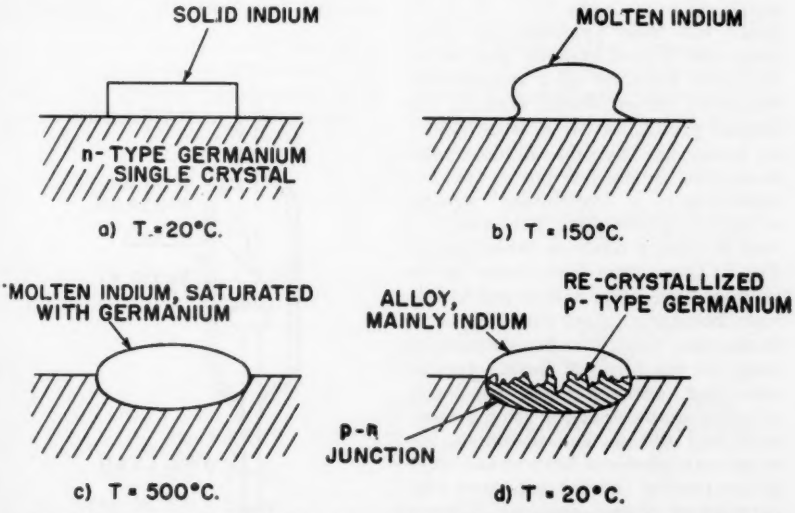


Figure 4. Steps in the formation of one form of alloy junction. After cooling, recrystallized p-type germanium forms a junction at the original solid-liquid interface.

due to a single polarity of charge to as high a degree as with actual p-n junction formation.

Plated metal contacts can also be used to obtain rectifying and transistor action but they share with the point contact some disadvantage with respect to single-polarity charge flow in the forward direction. With a recently described technique, using liquid jets to electrolytically etch and then (by reversal of battery polarity) to plate on the same area, transistors have been made with fair success (28).

Protection of finished transistors and diodes is a matter of great importance. As a result of experience to date, neither hermetic seals nor moisture-resistant plastic cases are a sufficient criterion for complete stability. Both can, however, be satisfactory and both are being used. Nevertheless, the method of encapsulation, and the treatment of the exposed semiconductor surfaces prior to encapsulation, are critical manufacturing factors which are not yet completely understood. For this reason, variability during life remains a problem on some transistor types.

PRESENT COMMERCIAL DEVICES

Today, there are already a substantial number of commercial devices using semiconductors. We have had selenium and copper oxide rectifiers for many years and these will not be mentioned further. Similarly, the thermistor, which consists of a small piece of semiconductor with two electrical leads, has been available for many years and is used in many ways as an electrical indicator of a temperature variation. Since World War II, the largest production of a semiconductor device has been the simple crystal diode, that is, the low-power rectifier consisting of a "catwhisker" against a piece of germanium. The construction of such a diode is illustrated in Fig. 5. These have been made by the millions both in the U.S.A. and in Europe. Somewhat more recently, other diodes have been introduced commercially in the form of small selenium cells and, perhaps more important, junction rectifier devices using germanium and silicon. The silicon diodes, made with junctions, have an extremely low reverse current compared with germanium diodes and this current does not increase to excessive values when the operating temperature is raised. Such units should have wide usefulness in switching applications where complete isolation is desired between the two terminals in the cutoff condition. Germanium junction diodes have been made with sufficient cooling to withstand large amounts of power. In electrical performance,

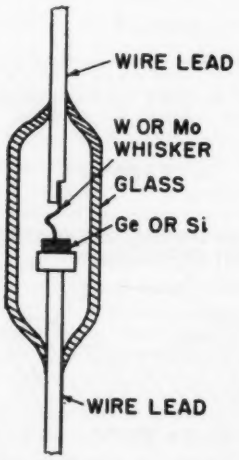


Figure 5. Construction of a germanium point-contact diode.

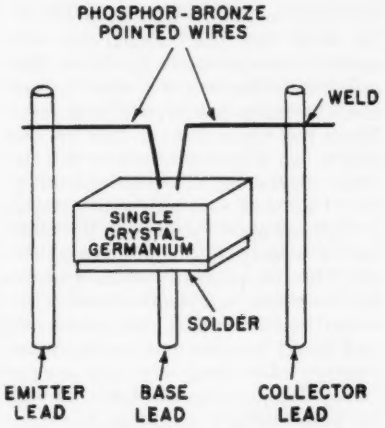
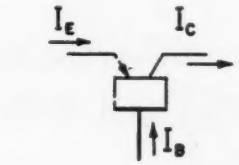


Figure 6. Construction of a point-contact transistor.

these power rectifiers are greatly superior to the selenium and copper oxide units hitherto obtainable. They have yet, however, to prove themselves from the point of view of long-time reliability.

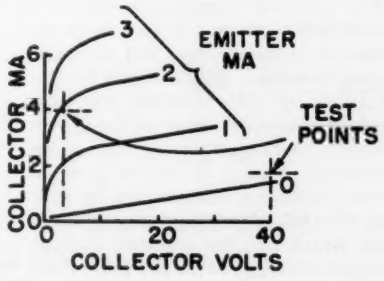
Turning to the transistor, the early forms were of the point-contact variety. The general construction of a point-contact transistor is shown in Fig. 6. Two point contacts are placed on the surface of a piece of germanium and it is found that the electrical current between one of these point contacts (collector) and the base of germanium is varied by the electrical current which flows into the second point contact (emitter). Best transistor action with point contact transistors is not achieved without the introduction during manufacture of a mysterious electrical process called "forming." In the process, an excess of current is passed through the collector contact to the n-type germanium. After this has been done, transistor action is greatly enhanced. Point-contact transistors vary in their characteristics when surface conditions of the germanium are altered and also have variations when point-contact pressures and positions are changed. The theory of the forming process is sufficiently obscure so that art, rather than science, must be used to make a practical device.

There are a number of commercial types of point-contact transistor available and Fig. 7 gives a composite characteristic. In contrast to electron tubes, point-contact transistors use voltages



$\alpha_{cb} = 2.5$
1kc GAIN = 20db
1kc NOISE = 50db
OSCIL. LIMIT = 5 TO 50 Mc.

Figure 7. Composite characteristics of commercially available point-contact transistors.



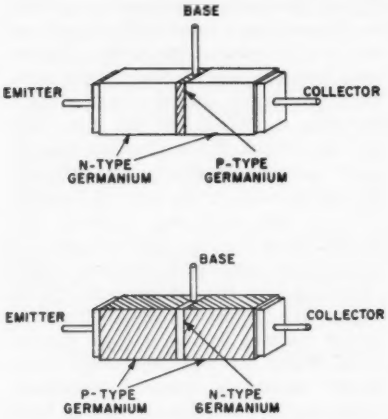


Figure 8. Construction of grown junction transistors.

about one tenth as high, that is, they operate with from 10 to 20 volts, usually at a few milliamperes. Power amplification is usually of the order of 100 times (20 db), and residual fluctuation noise is quite high compared with electron tubes. Commercial units will operate as high as 50 Mc. or more. Point-contact transistors are of particular usefulness in switching circuits and where the highest frequency of operation is necessary. For many ordinary amplifier applications, the point-contact transistor is not as reliable nor as stable in operation as junction transistors.

The junction transistor is the most recent semiconductor device to be introduced commercially, and it is expected to have a great impact on the electronic art. The junction transistor can be constructed either by the grown junction technique or by the alloy technique. Figure 8 shows how a grown junction is constructed by cutting off a bar of single crystal germanium which had the impurity content changed during growth. The other construction method, Fig. 9, uses two of the alloy junctions on each side of a wafer, as also shown in the figure. One can make either a p-n-p type of transistor or an n-p-n type of transistor. These two types differ in that the one makes use of hole conduction and the other makes use of electron conduction. One uses a battery with a negative potential on the crystal, the other uses a battery with a positive potential. The two polarities of the transistor, therefore, complement one another and can be used in very interesting and novel circuits not available for electron tubes which have only one polarity. The junction transistor, as with the point-contact transistor, has three leads. The current to be controlled comes in one lead (emitter) and goes out a second lead (collector) and is controlled by the voltage or current which is impressed on the third lead, the base. Since the entire unit is a single piece of crystalline germanium, the junction transistor looks like a small piece of material with three soldered connections.

Commercial junction transistors are made by both the grown technique and the alloy technique and in both p-n-p and n-p-n polarities. In general, these

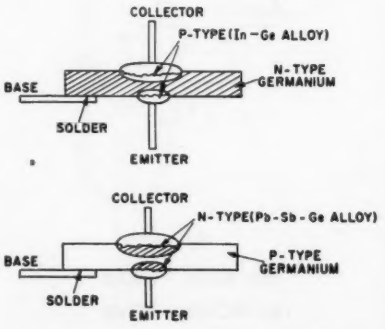


Figure 9. Construction of alloy junction transistors.

commercially available transistors have similar characteristics and one may consider them all as approximately equivalent. There are, of course, differences in the method of encapsulation, in the size, and in details of the currents and voltages for particular operating conditions. The power amplification at low frequencies is about 10,000 times, or 100 times greater than the point-contact unit. The noise at low frequencies is much less and compares favorably with electron tubes. The frequency performance of commercial junction transistors is not as good as point-contact types, perhaps a tenth as high.

Among present-day commercial semiconductor devices mention should be made of the photodiode, the phototransistor, and even the vidicon television pickup tube, because all these use semiconducting or photoconducting materials of the same general nature. Although these are outside the scope of the present paper, it should be kept in mind that the future may see a great increase in the importance of solid-state light-controlled devices in electronics.

PRESENT LABORATORY DEVELOPMENTS

So far, commercially available devices have been described. We will now consider devices which are still in the experimental stage in the laboratory.

The first important limitation of commercial transistors is their limited frequency response. There has been much work to overcome this and Fig.

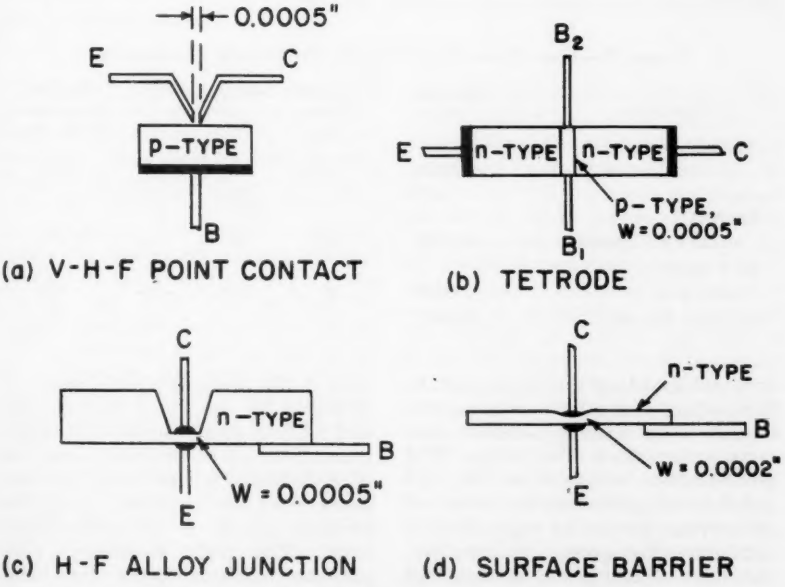


Figure 10. Four experimental high-frequency transistors.

10 shows four experimental developments. In the point-contact transistor, by use of very close point spacing, low resistance germanium of the *p*-type, it has recently been possible to attain an upper oscillation limit of over 400 megacycles (29). (See Fig 10 (*a*).) Unfortunately, the point-contact transistor is not easily applied to amplifier use.

A second development in improving frequency response is the grown junction *tetrode* transistor also shown in Fig. 10 (*b*). This consists of a very small grown junction arrangement with two base connections on opposite sides of the base layer (30). By placing a bias between these leads, transistor operation occurs only in a very small area near one of the leads. This reduces the base resistance and diffusion capacitance sufficiently so that high-frequency operation is improved.

A third improved high-frequency transistor retains the practical advantages of the alloy technique (31), as shown in Fig. 10 (c). Again one notes the use of extremely close spacing between the electrodes but, at the same time, the germanium wafer tapers into a thick section so as to retain very low base resistance. The junctions themselves are very small in area.

Still another development in high-frequency transistors is the one which has been called the "surface barrier" transitor (28), Fig. 10 (*d*). As with the other high-frequency types, this also makes use of low-resistance germanium, close spacing between the electrodes but, with this construction, the electrodes are plated rather than being metallic points or crystallized germanium.

Table II shows the approximate performance of the four types of high-frequency transistors as estimated from

TABLE II.—*Four Experimental High-Frequency Transistors.*

Transistor	Spacing, in.	Current Gain, a_{cb}	Amplification, decibels 0.5 Mc.	10 Mc.	Highest Reported Oscil., Mc.
V-H-F Point Contact	0.0005	—1.5	not suited	not suited	420
Grown Junction Tetrode	0.0005	6	32	20	130
H-F Alloy Junction	0.0005	40	38	14	75
Surface Barrier	0.0002	12	33	20	70

data published on the types (28-31). It is noticed that all the units use extremely close spacing between contacts or junctions. The *p*-type VHF point-contact transistor is not well suited for amplification but it tops all other transistors in its upper limit of oscillation. The grown junctions tetrode has excellent performance at high frequencies with some sacrifice, however, in low frequency performance as indicated by the lower current gain and reduced amplification at 0.5 megacycles. Because an extremely small bar of germanium is used and two base connections are required, the grown junction tetrode is not easily fabricated. The high frequency alloy junction transistor, on the other hand, is an extension of the more conven-

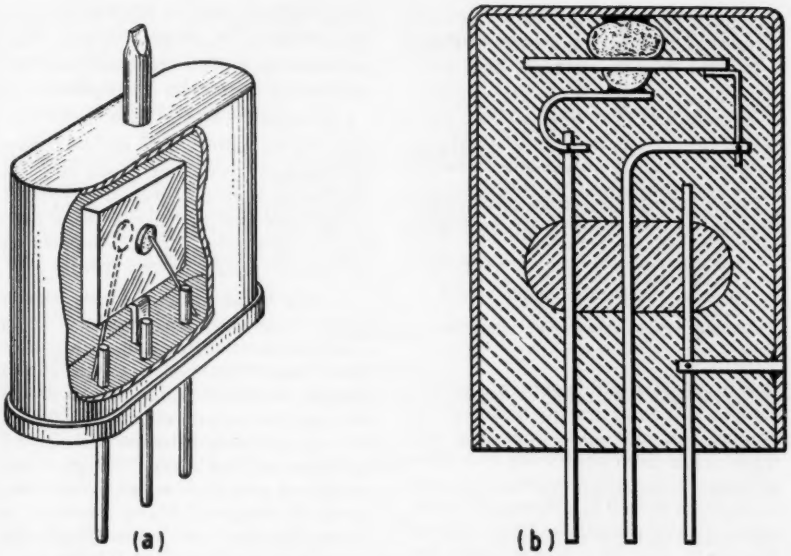


Figure 11. Two constructions for experimental power transistors, (*a*) liquid cooled, (*b*) conduction cooled.

tional alloy junction transistor which retains normal low frequency performance and yet operates well at high frequencies. Such a device, for example, could be universally used without sacrifice in voltage handling ability or performance. The surface barrier transistor makes up for the reduced quality of the plated emitter contact by an even closer spacing than the other types. As a result, it has reduced voltage handling ability and low power dissipation. As with the tetrode, there is some sacrifice in low-frequency performance in favor of high-frequency performance.

Another limitation of present commercial transistors is power handling ability. Fortunately, all the junction types of transistor are capable of operation at considerably higher current density values than normally necessary for small signal applications. The chief problem for a power transistor is dissipation of the additional internally generated heat. This can be done in many ways (32), of which the simplest are either liquid cooling, Fig. 11(*a*), or conduction cooling, Fig. 11(*b*). In the latter, the germanium wafer and/or its junction electrode is fastened to a piece of conducting copper which can radiate or conduct the heat to cooler environments. These particular units are experimental germanium ones which dissipate approximately one to three watts without auxiliary heat conduction from the metal envelope. By careful attention to the heat conduction means from the wafer to the envelope, and auxiliary cooling, considerably higher dissipations have been attained experimentally in the same size construction. Power junc-

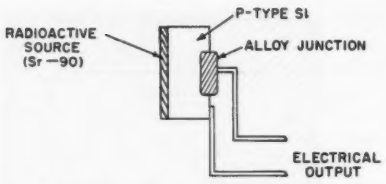


Figure 12. The electron-voltaic effect used to convert radioactivity to electrical power in a p-n junction.
Available radioactive power=4×10^{-3} watts/curie
Available radioactive current=6×10^{-9} amp/curie
Battery output current=10^{-3} amp/curie

tion transistors of this general type are capable of peak currents of about one ampere and maximum operating voltages of the order of 40 volts. In a Class B connection, wherein the transistor dissipation is held to a minimum, maximum power outputs of the order of five watts can be achieved with two units operated from a 12-volt supply.

Another laboratory development is the silicon junction transistor. Silicon single-crystal material of satisfactory quality has been available for some time and its application to alloy junction transistors required the development of suitable techniques. An alloy junction transistor using silicon has been developed which has current amplification factors of 20 or more and a power gain of between 35 and 40 db. at low frequencies. Such a transistor exhibits the expected freedom of changes in uncontrolled current with temperature; in fact, the uncontrolled current is so small as to be hardly detectable. Operation appears to be feasible at temperatures well in excess of 100° C. The perfection of presently available silicon crystals, and the techniques of surface treatment, have not reached the stage common with germanium and considerably better results are yet to be anticipated.

Laboratory results have been obtained with many other special forms of transistor but they have not yet reached as high a degree of practicality as those outlined herein. One may expect, however, further interesting and novel devices in the future.

It may be of interest to mention a recent laboratory development using a semiconductor junction in a rather novel way. This has been called an "atomic battery," and it makes use of the electron-voltaic effect (33). There have been many attempts to convert the electrical and kinetic energy of the process of radioactivity into some useful form of power. Many attempts to accomplish this have led to efficiencies of the order of less than 0.1 per cent. After a careful survey and experimental studies of many other conversion mechanisms, it was found that the electron voltaic effect in a semiconductor junction of silicon gave higher efficiencies and holds promise for considerable future improvement. The method is shown in Fig. 12. Beta particle emitting radioactive material is placed near an n-p alloy junction of silicon. The bombardment produces a low voltage at high current in the junction. The radioactive half life of the strontium 90 material is of the order of 20 years, but radiation damage to the semiconductor material must also be considered. Nevertheless, the future may see such sources of power, particularly in connection with transistorized equipment which operates on such extremely low powers.

THE FUTURE OF SEMICONDUCTOR DEVICES

As to the transistor, it will improve with respect to uniformity, reliability and cost. At the same time the frequency range will be extended, and

the power levels will increase to the point where even high-power industrial applications will be possible. Multi-electrode and special-function transistors will also be developed.

Many who have seen at first hand the rapidity with which the transistor has developed, believe that this device is but the first of many remarkable solid-state devices. The close relationship of photo effects and semiconduction can be expected to lead to useful improvements in all types of light-sensitive and light-producing devices.

The most basic steps, however, will come only as a result of improved materials. In this area, the chemist, the physicist, the ceramicist and the metallurgist will all find ample opportunity to contribute to the electronic field and to a degree not hitherto achieved.

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AUTOMATIC CONTROL FOR MARINE BOILERS

P. S. DICKEY and C. H. BARNARD

THE AUTHORS

Paul S. Dickey, vice-president of Bailey Meter Company, was graduated from Purdue University in 1925, and has been associated with the company since that time. The holder of numerous patents, he is a recognized authority on power plant and process control.

In 1953 Dickey received the honorary degree of Doctor of Engineering from Purdue University. He is a Registered Professional Engineer, a member of the American Society of Mechanical Engineers, Cleveland Engineering Society, the Society of Naval Architects and Marine Engineers, the American Society of Naval Engineers, and Sigma Xi, engineering research fraternity.

Clayton H. Barnard, staff engineer, Bailey Meter Company, was graduated from the University of New Hampshire in 1935. Associated with the company since his graduation, he was assigned to the Boston district office and the Cleveland Main Office application engineering department prior to his appointment as staff engineer. He is the holder of several patents.

Barnard is a member of the American Society of Mechanical Engineers, the Instrument Society of America, Cleveland Engineering Society, and the American Society of Naval Engineers. In 1951 he was awarded the A.S.M.E. Melville Medal and is author of numerous articles and papers.

Since the late 1930's, the application of automatic control to marine boilers has been increasing. Today practically all new merchant ships are equipped with automatic control, and its application to Naval combat and auxiliary vessels is growing rapidly. The objectives in the application of automatic boiler control are to insure the following:

1. Maximum fuel economy
2. Ability to follow variable loads
3. Safety and reliability under all conditions, including untenable machinery spaces
4. Minimum number of required operating personnel

Maximum fuel economy is obtained by maintaining steam pressure and temperature at the design conditions in order to insure maximum turbine performance, and by controlling the ratio of fuel and air within the furnace to maintain maximum combustion efficiency. The elimination of smoke by proper maintenance of the fuel-air ratio is inherently obtained along with maximum combustion efficiency, an essential under war-time conditions. Feed Water Control capable of properly controlling the feed water under all conditions, automatic steam temperature control on boilers equipped for temperature regulation, and auxi-

liary controls of various types contribute to the overall fuel economy of the ship.

The boiler control equipment must be capable of following fluctuating loads over rather wide ranges, at the same time maintaining maximum fuel economy and a minimum of smoke. With this function automatic control can properly regulate the boilers under maneuvering, as well as steady steaming conditions, with a minimum of attention from the operators. The ability of the controls to handle wide ranges in power is limited by the range of the auxiliaries which must be controlled, and the proper selection of burners and auxiliaries is a necessity if the best installation is to be obtained.

Safety and reliability of operation are achieved inherently with a properly maintained control system, since it is on the job continually, every second, and under battle damage conditions will continue to operate for extended periods of time without any operator attention whatsoever. At times when the machinery space becomes untenable the boiler control will continue to provide power for the ship's operation. During the last war, more than one vessel was saved from destruction by the ability of feed water regulators to maintain the water in the boiler drum, automatically, under extreme battle conditions.

The number of men necessary to operate marine boilers can be appreciably reduced by the proper application of automatic control equipment. With the great majority of the functions handled automatically, the operators become supervisory in nature, and a limited number of men can intelligently supervise a large amount of equipment. At the present time, merchant type vessels are operating the firerooms with two men per watch.

In the case of Naval vessels, the release of a number of men from the machinery spaces can reduce the complement of the ship, or permit the released men to be utilized elsewhere to increase the combat effectiveness. Where the complement is reduced, savings can be made in ship design because of reduced requirements for sanitary facilities, stores, refrigeration, evaporators, and auxiliary generators. Existing Naval vessels having automatic control have recently been instructed to reduce the number of men on steaming watches to take more complete advantage of the controls.

RELATED DEVELOPMENTS IN MARINE POWER PLANTS

Since the end of the last war, there have been a number of related developments in the marine power field which extend the use of automatic control. Some of these developments are as follows:

1. Increased propulsion power
2. Increased steam pressure and temperature
3. Increased boiler and turbine ratings
4. Improved fuel burners — wide range
5. Improved auxiliaries
6. Improved operator training
7. Improved utilization of space

The trend in the field of ship propulsion in the United States has been toward increased propulsion power, which requires larger boilers, turbines, and auxiliaries. Nearly all of the post-war designs for cargo ships and tankers call for speeds of 18 knots and up, and passenger vessels in excess of 30 knots. The increased propulsion power has of course brought about an increase in the consumption of fuel, so that maximum fuel economy is more important

than ever. With automatic control capable of handling fluctuating loads, a minimum of personnel can handle the ship with complete safety and reliability, despite the increased power.

Increased steam pressures and temperatures are employed in the interest of greater overall fuel economy. The higher pressures and temperatures require closer regulation, since the actual operating conditions get closer to the safe operating limits as the pressures and temperatures increase. Thus, improved operation is required and this can best be handled by suitably designed automatic control.

While propulsion power, steam pressure, and steam temperature have been increasing, the relative size of boilers and turbines has not increased in proportion, with the result that there is less energy storage in the boilers and turbines than heretofore, which requires faster manipulation of burners, blowers, pumps, and other auxiliaries to compensate for the reduced storage. This requires control designed for a high rate of speed.

Improved fuel burners operating over a wide range have somewhat simplified the problem of fuel oil control, since it is no longer necessary to cut burners in and out under manuevering conditions. Burners capable of handling load ranges of 10 to 1 and greater are now available from several manufacturers and permit complete automatic operation over the normal operating range.

Improved auxiliaries have been developed: blowers, operating at high speeds, deliver large quantities of air at adequate pressure; feed water pumps, smaller in overall size, deliver large quantities of water at adequate pressure; and other auxiliaries such as fuel oil pumps, condensate pumps and the like have been designed for improved performance. The necessity of equipping these auxiliaries with suitable control devices in order to take fullest advantage of their design features is of paramount importance.

During the past few years, the Naval Boiler and Turbine Laboratory at Philadelphia has inaugurated courses in automatic control for Naval personnel, and major manufacturers of control equipment have established schools which Naval personnel may attend without cost. The manufacturers also conduct schools which may be attended by Merchant Marine or Navy personnel. This increased and improved operator training is providing a pool of men in the Merchant Marine and in the Navy capable of properly maintaining automatic control equipment, so that its usefulness may be utilized to the utmost.

Improved utilization of space has permitted concentrating all control functions at a common location, which allows the use of a minimum of personnel, and at the same time greatly increases the operators' knowledge of all factors relating to the operation of the plant. The day of the completely remote controlled marine propulsion plant is practically here.

AUTOMATIC CONTROL SYSTEMS

Automatic control has been applied aboard ship for control of combustion, boiler feed water, steam temperature, evaporators, feed water pumps and heaters, fuel oil pumps and heaters, and various other auxiliaries. Properly designed controls can handle these functions with only casual operator supervision under all normal operating conditions.

In general, three types of control media have been employed aboard ship, but for boiler control the pneumatic system is preferred. Pneumatic

systems in general have more than adequate speed for control of marine boilers and auxiliaries, and by the simple expedient of utilizing air at relatively low pressures, and utilizing control air compressors operating at a somewhat higher pressure, a means of providing storage for an appreciable period of time is obtained. A moderate size storage tank can provide sufficient air for normal operation for perhaps ten or fifteen minutes, during which time the operators can be warned of compressor failure and take the necessary steps to place another compressor into operation, connect to the ship's service air system, or place the control equipment on direct manual control. Since the air storage tanks are of relatively low pressure and simple construction, they are inexpensive and do not require any appreciable amount of space. Electric control can be lost without warning, and hydraulic controls, which usually operate at much higher pressures, require large and expensive storage tanks to provide a reserve in the event of failure.

Another advantage of pneumatic control over a hydraulic system is that no special fluids are required to reduce fire hazards in the event of a leak, and the control medium itself cannot be lost through leakage. There is always ample air available, even though leaks may occur.

SPECIAL FEATURES FOR MARINE APPLICATIONS

The application of control equipment to shipboard use introduced requirements not found in shore installations, which necessitated design changes to make equipment suitable. A shore based installation is in all respects stationary, while a ship is subject to severe pitch and roll. Consequently, suitable automatic control components have been designed to operate satisfactorily under all conditions of pitch, roll and trim. Considerable variations in ambient temperature are experienced in many shipboard installations, and automatic control equipment capable of operating at ambient temperatures from 40 F to 150 F is now readily available.

For Naval combat vessels, the Bureau of Ships requires shock tests in accordance with Specification MIL-S-901, and vibration tests in accordance with Specification MIL-T-17113. These tests are designed to equal or exceed the shock and vibration which may be imposed on equipment during the normal life of a combat ship, and for a time these requirements taxed the ingenuity of automatic control designers. However, by co-operation between the manufacturers, the Naval Research Laboratory and the Naval Boiler and Turbine Laboratory, suitable control equipment has been developed, tested for shock and vibration, and proved for performance, so that equipment is now available, at moderate cost, to meet all of the requirements. It is interesting to note that many of the design changes made to meet shock and vibration requirements improved the quality of the control components to such an extent that they were adopted universally, thus increasing the quality of control equipment for all applications.

TYPICAL INSTALLATIONS

Over one thousand merchant vessels have been equipped with combustion control, the majority of them tankers or one of the Type C Maritime Commission cargo designs. A typical installation is shown in Figure 1. The basic control arrangement for these boilers is illustrated in Figure 2, and consists

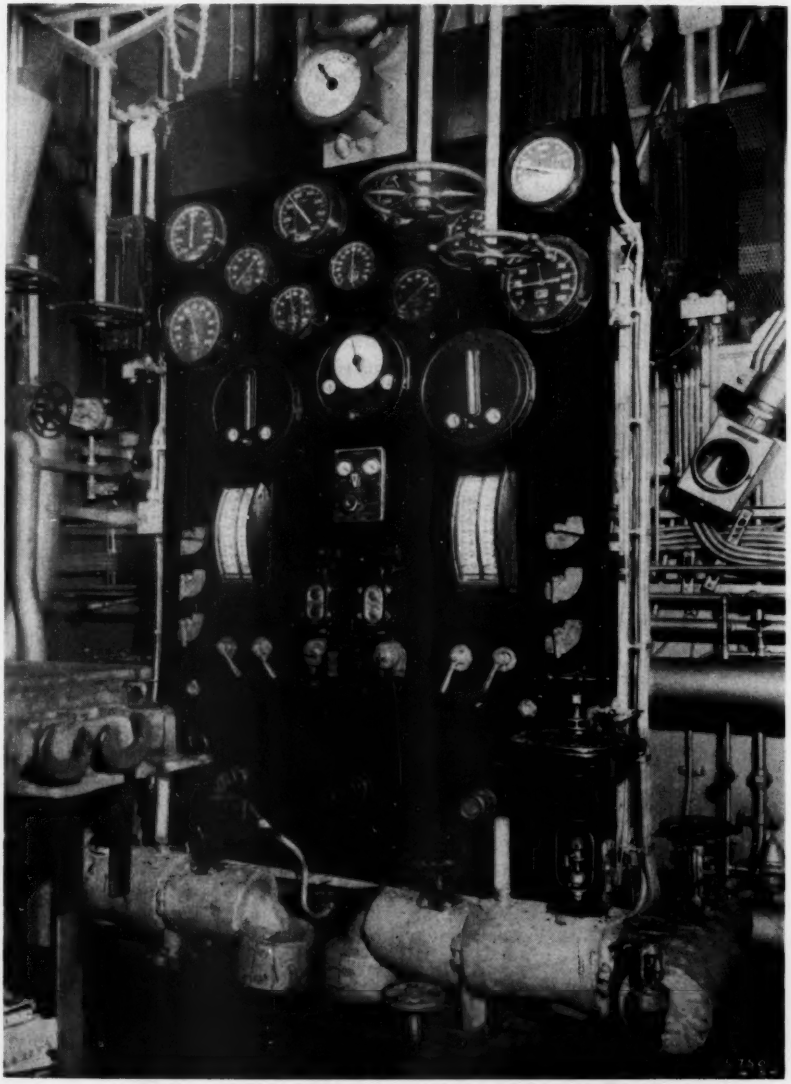

Figure 1. Boiler Control Panel on Typical Maritime Commission Tanker.

of a Master Steam Pressure Controller which operates through a relay and hand-automatic selector valve to regulate the fuel oil to the burners, and through a second relay and hand-automatic selector valve to regulate the position of the forced draft dampers. An Oil Pressure-Air Flow Ratio Controller is installed to control the fuel-air ratio by balancing the oil pressure at the burner tips against the drop across the air registers, and introduce a signal to the averaging relay to readjust the forced draft damper to maintain the proper fuel-air relationship. Boiler drum level is measured by an indicator-controller, which operates through a relay and hand-automatic selector valve to regulate a control valve in the feed water line. This control is of the simplest type, and may be applied only to boilers having the simplest type of auxiliaries. However, for the purposes for which it is normally used it is entirely adequate and meets the major objectives of an automatic boiler control system.

ATLANTIC REFINING COMPANY TANKERS

The Atlantic Refining Company has pioneered in the development of propulsion plants for tankers, leading the industry in higher steam pressures, temperatures and improved auxiliaries. Figure 3 illustrates the boiler control used on this type of ship. A Master Steam Pressure Controller operates through suitable relays and hand-automatic selector valves to

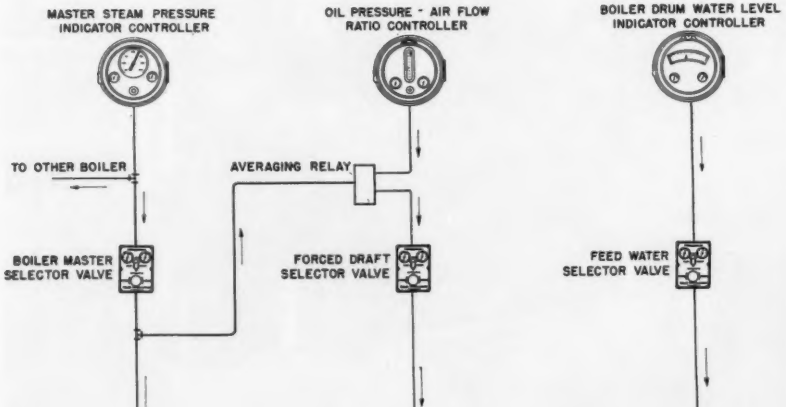
MASTER STEAM PRESSURE INDICATOR CONTROLLER
OIL PRESSURE - AIR FLOW RATIO CONTROLLER
BOILER DRUM WATER LEVEL INDICATOR CONTROLLER
TO OTHER BOILER
AVERAGING RELAY
BOILER MASTER SELECTOR VALVE
FORCED DRAFT SELECTOR VALVE
FEED WATER SELECTOR VALVE
FUEL OIL CONTROL VALVE
FORCED DRAFT DAMPER CONTROL DRIVE
FEED WATER CONTROL DRIVE

Figure 2. Schematic Control Diagram, Maritime Commission Tanker.

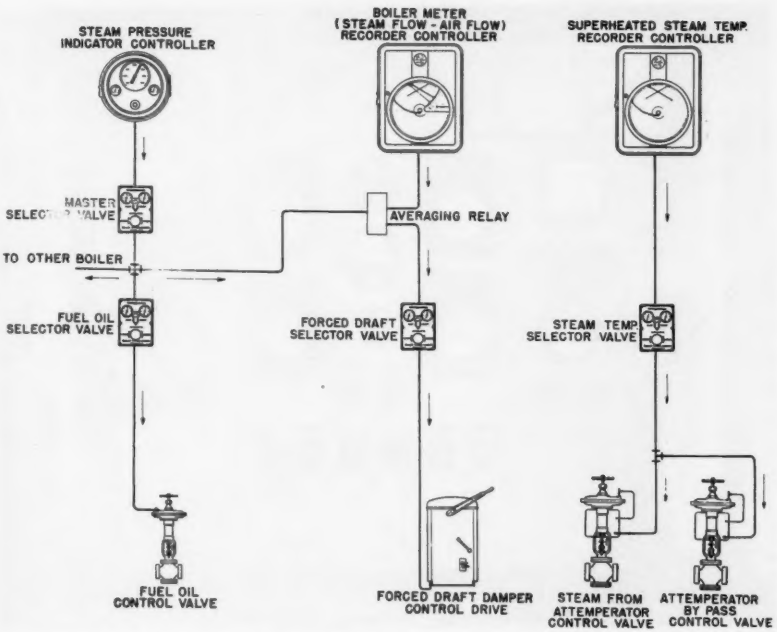
STEAM PRESSURE INDICATOR CONTROLLER
BOILER METER (STEAM FLOW - AIR FLOW) RECORDER CONTROLLER
SUPERHEATED STEAM TEMP. RECORDER CONTROLLER
MASTER SELECTOR VALVE
AVERAGING RELAY
TO OTHER BOILER
FUEL OIL SELECTOR VALVE
FORCED DRAFT SELECTOR VALVE
STEAM TEMP. SELECTOR VALVE
FUEL OIL CONTROL VALVE
FORCED DRAFT DAMPER CONTROL DRIVE
STEAM FROM ATTEMPERATOR CONTROL VALVE
ATTEMPERATOR BY PASS CONTROL VALVE

Figure 3. Schematic Control Diagram, Atlantic Refining Company Tanker.

regulate the fuel oil valves supplying oil to the burners, and through a second relay to regulate the forced draft damper. Rather than an oil pressure-air flow ratio instrument, a Boiler Meter, utilizing measurements of steam flow and air flow, similar to that used universally on stationary boilers, provides a corrective signal through the averaging relay to maintain the proper fuel-air ratio by readjustment of the forced draft damper. Steam temperature control is obtained by measuring the outlet steam temperature, and regulating a 3-way flow control valve which by-passes part of the steam through a submerged coil in the mud drum of the boiler. Feed water control is of the 3-element type, utilizing steam flow, water flow, and drum level measurements in order to provide a water flow approximately proportional to the steam output, and modified from level within the boiler drum. This type of feed water control is growing in usage very rapidly, as the steam release per square foot of water surface is rapidly increasing as higher capacities are obtained from given drum sizes. The control panel is shown in Figure 4.

The Atlantic Refining Company has also pioneered in the use of recording instruments for critical services, and their experience has proven the wisdom of their decision. A continuous record is made of each day's operation, and this record provides a means of detecting deviations from normal operation so that corrective steps may

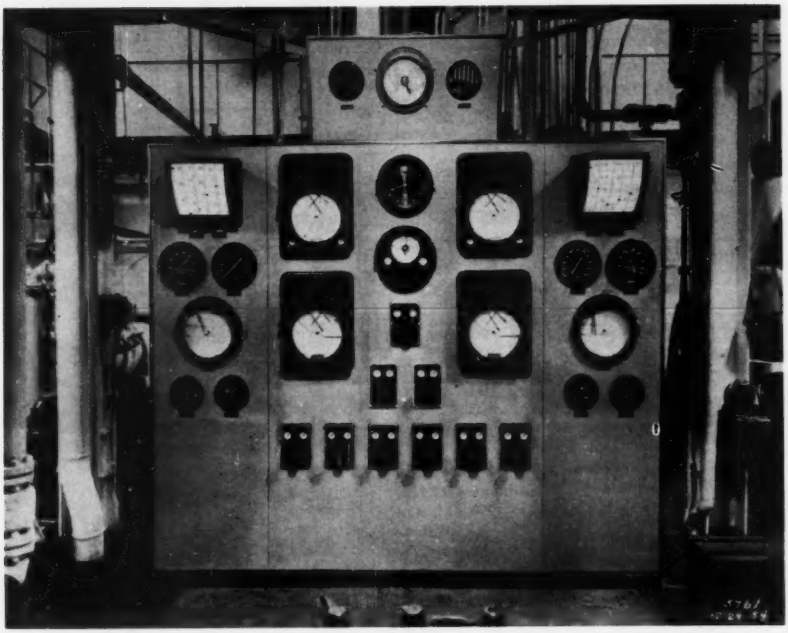

Figure 4. Boiler Control Panel on Atlantic Refining Company Tanker.

be taken to remedy the trouble before a ship outage is required.

Where an indicator gives only an instantaneous reading, a recorder shows simply and graphically the direction and rate of change as well. This added information is of great value to the operators, and assists them in getting the utmost from the machinery.

While recorders have been used sparingly aboard ship, all stationary plants have found them essential, and with improved design recorders which are better suited for shipboard use now available, their use will increase.

NAVAL COMBAT VESSELS

For many years the Navy has been interested in the application of automatic control to marine boilers on combat ships, but the full utilization of control was delayed pending the development of suitable fuel burning equipment and auxiliaries necessary to operate over the required wide range in power. Just prior to the last war the USS *Dahlgren,* a World War I four-stack destroyer, was converted as an experimental ship, but saw considerable active service on convoy duty. This ship was reboilered with modern high pressure boilers, and the installation included automatic boiler control equipment. Valuable experi-

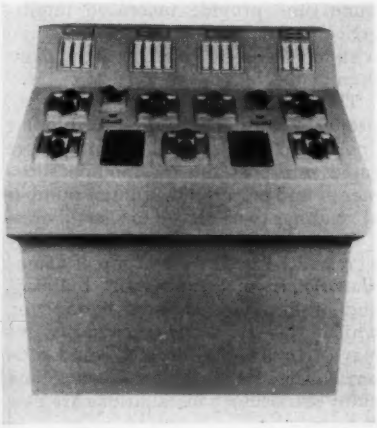

Figure 5. Boiler Operating Console for USS *Northampton*, CLC-1. Includes complete control for steam pressure, steam temperature, fuel-air ratio, and feed water.

ence was gained in the installation, and the experience with this ship was of considerable value to the Navy and manufacturers when the war was over and the Navy had time to consider future developments.

Shortly after the war the USS *Robinson*, DE-220, was equipped with automatic boiler control to gain further experience in controlling Naval type boilers under actual operating conditions. Moderate success was experienced with this installation, but the results indicated that further developments in controls, oil burners and auxilaries were required. Up until this time control equipment supplied for Naval vessels had been adaptations of commercial equipment, and it was agreed that this equipment was too large, did not lend itself to easy installation in crowded fire rooms, and was not suitable for shock and vibration. Accordingly, it was decided that an entirely new concept in control components was required for Naval service, which would permit an installation of minimum space and weight, with an operating center providing all necessary functions in a very small space, and with all components suitably designed for shock, vibration, pitch and roll, and other special requirements. An extensive study indicated that a pneumatic control system was preferable, utilizing locally-mounted transmitters, which would send pneumatic signals proportional to the measurements to a small console upon which would be located small size indicators for the various measured functions, and reduced size control stations. This concept proved sound in many ways, as the transmitters could be mounted locally without taking undue space, and a single small pneumatic transmission line was connected to the control center. This eliminated the necessity of running high pressure piping into the control center, which contributed to the safety of the installation and yet provided adequate indications for all control purposes.

A study of the indicators to be mounted on the control console showed that a small vertical scale gage gave greater readability in a small space, and by stacking these gages side by side a "pointer pattern" could be established so that the operator could determine normal operations by a casual glance at a group of related measurements rather than by comparing individual readings of numerous gages.

The first installation of this type was made on the USS *Northampton*, CLC-1, a task fleet command ship built on a *Baltimore* Class hull, and equipped generally with *Baltimore* Class machinery. Prior to the installation aboard the *Northampton* the equipment was thoroughly tested at the Naval Boiler and Turbine Laboratory and the Naval Research Laboratory. A complete operating console for one boiler is shown in Figure 5. Although

this is a two-furnace boiler requiring separate control of fuel and air to each furnace, the control console is remarkably small in size, yet operating experience over 18 months indicates that the basic concept is entirely sound, and the operators can handle the machinery in this plant with a minimum of effort.

The equipment aboard the *Northampton* was installed in a very small space in an existing fire room, but because of the advanced design of the components and the console proper, the installation is entirely workable in every respect.

In the future, the concept of pneumatic transmitters for all measured functions will be carried to merchant as well as Naval vessels, because of the improved performance and ruggedness of the components, and the greater flexibility of application. The control centers will become small, and at the same time provide increased intelligence to the operator, and provide him with more absolute control of the plant than he has had in the past.

The use of small vertical scale gages, properly designed for developing "pointer patterns," will permit a more rapid interpretation of the measurements, and permit a maximum number of readings to be placed in a small area.

Control consoles of this general design will be used on a group of Canadian destroyer escorts now building, and others. This class of equipment, which was developed primarily for Naval service, has already found wide application in stationary plants, and many satisfactory installations are now in service.

Automatic control for Marine boilers has come a long way in the past 15 years, and another 15 years will find it universally utilized in all Marine propulsion plants.

STRESS CONCENTRATION FACTORS

ACKNOWLEDGEMENT

Theoretical stress concentration factors cannot in general be applied directly to strength values in practice, because of differences that exist between the hypothtical material assumed by theory and the crystalline structure used by the engineer. This article, written by Dr. R. B. Heywood, discusses some of the causes for this discrepancy and comments on methods of relating theoretical and practical factors. Prepared as a paper for the 1954 International Colloquium on Photoelasticity and Photoplasticity in Brussels, the article was published in the 4 February 1955 issue of *Engineering*.

The application of mathematical and experimental methods of stress analysis has produced a wealth of information on values of stress concentration factors for widely differing designs. The information has been correlated in the literature, and the designer is usually able to ascertain the value of the factor for the particular case in which he is interested. The maximum stress is then obtained by multiplying the value of the factor by the nominal stress. This knowledge of maximum stress can be related to the mechanical properties of the material, and so an estimate of strength can be made. In general, this estimate is not a reliable one, because the ideal homogeneous and elastic material assumed in arriving at the theoretical stress concentration factor does not accurately represent the structural material actually used in practice. This causes observed strength values of notched parts to be equal to or greater than is indicated by a direct application of theoretical stress concentration factors. If appreciably greater, then clearly a conservative and uneconomic design is produced when full allowance for stress concentration is made. However, a complete lack of consideration of the effect of stress concentrations can have, and indeed has had, a disastrous effect on the estimation of strength in certain instances. What is required is the use of some relationship connecting theoretical and effective factors, which not only produces realistic results, but is also of a sufficiently simple form for it to be employed in the design office without difficulty.

A number of relationships have been suggested, and in this paper an assessment of their qualities is made. The more complicated relationships, such as those requiring two or more notch sensitivity constants of the material, will not be considered, as they are not likely to be used in practice, even though they may have a more satisfactory theoretical basis for their derivation. For ductile materials, theoretical stress concentration factors

have not been applied successfully to estimates of static strength of notched parts. Plastic flow largely nullifies the effect of the stress concentration, so making this case of but little importance. One can merely ascertain the nominal stress at which yielding commences at the stress concentration, but this knowledge is of little practical value. For brittle materials, however, the theoretical solution is of considerable value in indicating strength, especially if allowance be made for size effect. Perhaps the most important application of theoretical results is the estimation of strength of parts subjected to fatigue, and it is particularly some aspects of this problem which will be considered.

THEORIES OF FAILURE INVOLVING THREE PRINCIPAL STRESSES

For unnotched parts, failure may not depend wholly on the maximum stress, but on some combination of the three principal stresses or strains acting at a point. Many theories of failure based on combined stress have been proposed[1], and in particular the maximum shear energy or Mises-Hencky criterion of failure has often given good agreement with experiment for ductile materials. According to the Mises-Hencky criterion, failure occurs when the stress f exceeds a critical value in the relationship

$$f = \frac{1}{\sqrt{2}} \sqrt{(P - Q^2) + (Q - R)^2 + (R - P)^2},$$

where P, Q and R are the principal stresses. However, when applying these theories to the determination of strength of notched parts, difficulties arise which show that solutions may still be inaccurate. For example the pointed V groove with zero root radius possesses an infinite stress concentration factor (s.c.f.), a value which would be still obtained if a combined stress theory were applied, in contrast to the finite value found in practice. This effect persists with V-grooves having a small base radius, when discrepancies between effective and theoretical factors would be finite but large.

Yen and Dolan[2], in reviewing the subject of notch sensitivity, have pointed out that s.c.f's. based on the shear energy criterion give only slightly lower values than those based simply on the maximum stress. For a grooved bar in tension and for a Poisson's ratio of 0.3, it may be shown that the effective factor ranges from 89 to 100 per cent of the theoretical factor for different sizes of groove. Most other theories of failure give percentages greater than 89, and so these laws can only partly account for the much larger discrepancies that are often obtained between theory and practice. A plate containing a free hole is an example where combined stress laws are of no use. In whatever manner the part be loaded, only the circumferential principal stress has any magnitude at the critical corners of the hole, and therefore all theories of combined stress must make the effective factor equal in value to the theoretical factor. Such a behavior is not usually obtained in practice. A further difficulty in accepting a combined stress theory as the only cause of discrepancy is that no differentiation in the notch sensitivities of different materials can be made—all materials must behave similarly. Thus the application of theories of failure based on combined

stress can modify values of the theoretical s.c.f. to give slightly closer estimates of strength, but nevertheless this accounts for only part of the correction required.

Influence of Plastic Flow.—In the static testing of notched ductile members the influence of plastic flow in modifying stress distributions is so great that a knowledge of theoretical stress distributions is of little value in indicating strength. Plastic flow or inelastic strain may also occur in fatigue, but to a much lesser extent. In fatigue tests on plain specimens in bending, Forrest and Tapsell[3], at the National Physical Laboratory, have shown that inelastic strain may cause a redistribution of stress during each cycle, so that the maximum stress becomes less than the calculated elastic value. This gives one reason why the fatigue strength in bending is usually greater than in axial loading.

A similar behavior would be expected with notched parts in fatigue, causing the maximum stress to be less than the elastic value; see Fig. 1. Inelastic strain is produced in both plain and notched specimens, but for direct loading stress redistribution can only occur in the latter case, so giving an effective factor lower in value than the theoretical s.c.f. Redistribution of stress is probably independent of size,

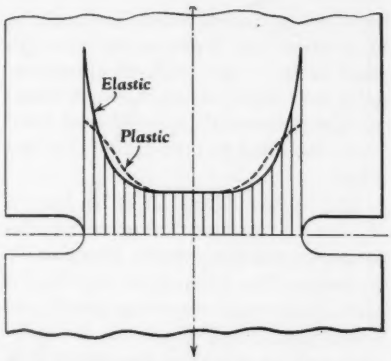

Figure 1. Redistribution of stress due to plastic flow.

so that the effective stress concentration factor (strength reduction factor) can be expressed simply as some function of the theoretical factor. The notch sensitivity index, to be discussed later, was originally devised by Thum to assess behavior under these conditions.

The importance of inelastic strain on notched fatigue strength has not yet been fully assessed: it will obviously have a greater influence with the more ductile materials, with increase in temperature of test, and with reduction in cycles to failure. However, inelastic strain probably has very little influence on the fatigue limit of

Figure 2. Effect of flank-angle term in Neuber's formula.

notched specimens which are made in a medium or high-tensile strength steel or in a high strength aluminum alloy and tested at room temperature, so that theoretical factors do not need to be modified to take account of this effect.

Size Effect.—Size effect is known to have a marked influence on the strength of notched parts, particularly in fatigue. The behavior is such that a part containing a very large notch possesses small stress gradients, and consequently material in the vicinity of maximum stress behaves as in a plain bar, and so produces the full reduction in strength as indicated by the theoretical s.c.f.* At the other extreme an exceedingly small notch could be of a size comparable with that of inherent imperfections within the engineering material, and then the notch would produce no further weakening effect than is already present, so making the effective s.c.f. of unit value. Thus the effective s.c.f. ranges in value from unity for the very small notch to that of the theoretical s.c.f for the large notch, but never exceeds this latter value. Intermediate values must necessarily be obtained for notches of intermediate size. Size effect can be appreciated most readily with cast iron, where large flaws due to the presence of graphite flakes normally make the material insensitive to notches. If, however, the size of notch is large compared with the size of graphite flakes, even this material would become notch sensitive. As size effect and the associated stress gradient has such a strong influence on the application of theoretical results to practice, it is suggested that when investigations are made to determine values of the theoretical s.c.f., consideration be given also to the measurement of the stress gradients and the second (and third) principal stresses at the point of maximum stress. Ultimately this would enable a more rational assessment of the different types of notches to be made.

Notch Sensitivity Index. — The above simple analysis on size effect demonstrates that it is not advisable to use the so-called notch sensitivity index q, defined as

$$q = \frac{K_f - 1}{K - 1}, \quad \ldots\ldots\ldots\ldots (1)$$

or

$$K_f = 1 + q(K - 1),$$

where

K_f = effective s.c.f. or strength reduction factor

$$= \frac{\text{stress causing failure in unnotched part}}{\text{nominal stress causing failure in notched part}}$$

and

K = theoretical s.c.f.

$$= \frac{\text{maximum stress in notch}}{\text{nominal stress based on minimum notched section}}$$

The terms on the right-hand side of this second formula are independent of size, and so no allowance is made for this effect on values of the factor K_f. Size effect could only be allowed for by making the value of the index q vary with size, but this would lead to confusion in the comparison of notch sensitivities of materials and in the application of theoretical results to practice. Thus at best, this well-known formula can only take account of the effect of plastic flow in modifying stress distributions.

* The assumption of a linear stress-strain curve is involved here.

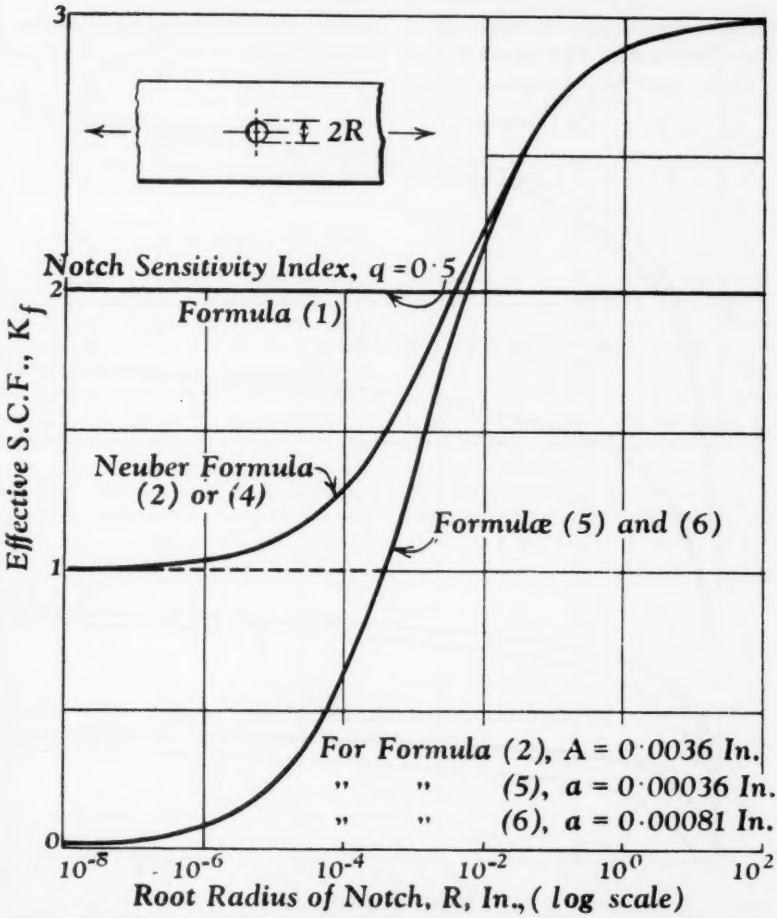

Figure 3. Notch size effect of transverse holes in specimens as assessed by various formulae.

NEUBER'S FORMULA

The difference between theoretical and practical factors has been explained by Neuber[4] on the grounds that the average stress acting over a small elementary block of material is significant, and this led him to suggest the empirical relationship

$$K_f = 1 + \frac{K-1}{1+\frac{\pi}{\pi-\omega}\sqrt{\frac{A}{R}}} = \frac{K+\frac{\pi}{\pi-\omega}\sqrt{\frac{A}{R}}}{1+\frac{\pi}{\pi-\omega}\sqrt{\frac{A}{R}}} \quad \ldots\ldots\ldots\ldots (2)$$

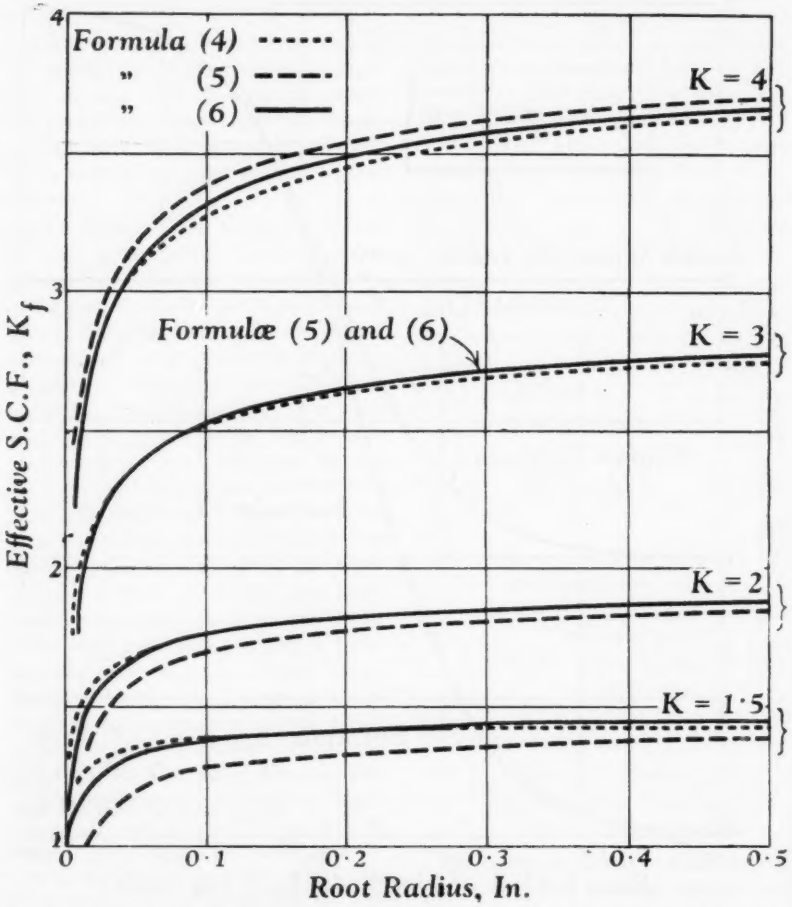

Figure 4. Comparison of notch size effect in fatigue for geometrically-similar specimens made of mild steel.

where

A = a material constant, having dimensions of length, and representing half the width of the elementary block,

R = root radius of notch,

ω = included angle of notch in radians, for notches made in the form of a V groove $\left(\text{flank angle} = \frac{\omega}{2}\right)$.

This formula satisfactorily takes into account size effect by incorporating a critical dimension of the notch, namely, the root radius, R. With pro-

gressive reduction in radius the effective factor becomes a smaller fraction of the theoretical factor, and so is in accordance with observed trends. In this formula the effect of flank angle is assessed twice, first by a dependency of the value of the theoretical factor on flank angle, and secondly by the term $\frac{\pi}{\pi - \omega}$. Both reduce the value of the effective factor with increase in flank angle. In order to verify this double influence of flank angle, accurate values of the theoretical factor for different flank angles and very accurate fatigue test data would be required. As far as is known precise information on both points is not available, and hence there is some doubt whether allowance for flank angle has been correctly represented in the formula.

The effect of flank angle correction can be appreciated in a particular case if a semi-circular notch is compared with a 120-deg. V-groove, both of which have the same value of the root radius of $R = 0.01$ in., and the same theoretical factor of $K = 3$ (obtained for the 120-deg. notch by using suitable other dimensions such as notch depth), see Fig. 2. By assuming a typical value of Neuber's material constant of $A = 0.0036$ in. (as given by Kuhn and Hardrath[5] for a 100,000 lb. per square inch steel tested in fatigue), then by formula (2) the value of the effective factor K_f is 2.25 for the semi-circular notch but only 1.71 for the 120-deg. notch. This difference in values is greater than would reasonably be expected, since the equality in theoretical factors and root radii must produce approximately equal stress

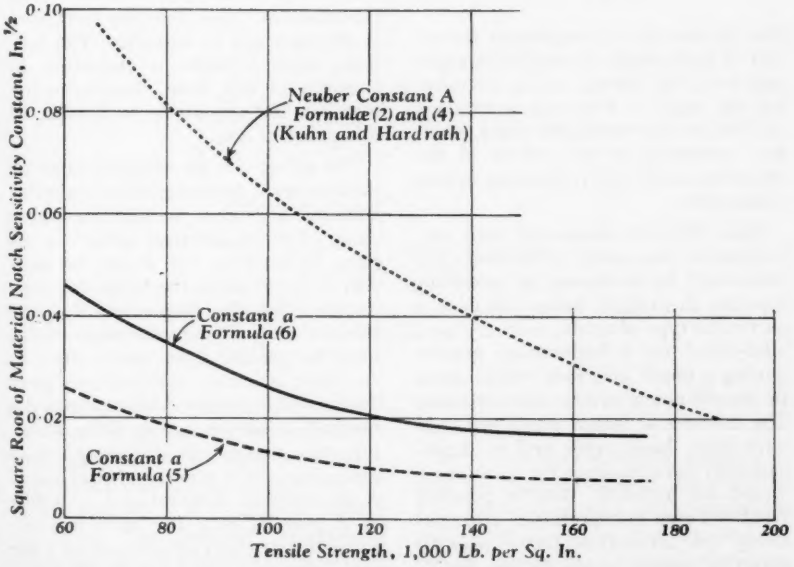

Figure 5. Approximate values of the material notch-sensitivity constants for steels.

gradients at the points of equal maximum stress. The practical factor for the semi-circular notch is not affected by flank angle, since the angle is zero, and so it follows that the factor for the V-groove must be low.

This comparison can be extended to cover any one of a series of U-grooves with a corresponding one of a series of V-grooves, so that for each pair the values of the quantities K, R and A are the same. In each case differences are obtained in the practical factors which can only be attributed to low values for the V-groove due to the influence of the flank-angle term.

If formula (2) is re-written in the form

$$K_f = \frac{K\sqrt{\frac{R}{A}} + \frac{\pi}{\pi - \omega}}{\sqrt{\frac{R}{A}} + \frac{\pi}{\pi - \omega}}, \quad \ldots\ldots (3)$$

then in the above comparison the effect of flank-angle correction is apparent from the second terms by varying the angle ω. Practical factors are reduced by increasing the angle, which for constancy in the values of the quantities K, R and A does not appear reasonable.

This difficulty associated with correction for flank angle in Neuber's formula may be overcome by assuming that the theoretical factor refers to a particular type of notch, such as a parallel-sided or a hyperbolic groove, having a depth and root radius equal to the V-notch under consideration. The theoretical factor would not then vary with flank angle, and so duplication in the allowance for flank angle would be avoided. This is possibly Neuber's own interpretation. Alternatively the theoretical factor may be given its normal interpretation for V-notches, and the flank-angle term could then be omitted, so giving the relationship

$$K_f = \frac{K + \sqrt{\frac{A}{R}}}{1 + \sqrt{\frac{A}{R}}} \quad \ldots\ldots (4)$$

This formula was given by Neuber in a preliminary stage of development of formula (2), and appears to be the more satisfactory alternative, as no restrictions are placed on the interpretation of the theoretical factor.

Neuber's formula (2), and also formula (4), possesses the further difficulty that the trend in size effect does not appear to be in accordance with what would be expected for notches of extremely small size. Engineering materials possess flaws and irregularities in their structure, and if an exceedingly small geometrical notch is introduced which is even smaller than the inherent flaws, then no reduction in strength can be expected. The formula does indicate a reduction in strength in this case, demonstrating that it is not applicable to notches of this order of size.

The effect can be demonstrated by considering geometrically - similar notched specimens having the same value of the theoretical factor for all sizes. In formula (2) it can be seen that however small the notch (as represented by the dimension R), the value of the effective factor must always be greater than unity—only in the limit is unity obtained—so producing some reduction in strength. An example is shown in Fig. 3 for a circular transverse hole, assuming a theoretical factor of 3 and a typical value of the material constant of $A = 0.0036$ in. A hole of only 10^{-7} in radius reduces the strength by as much as 1 per cent, even though inherent flaws of the order of at least 100 times this size

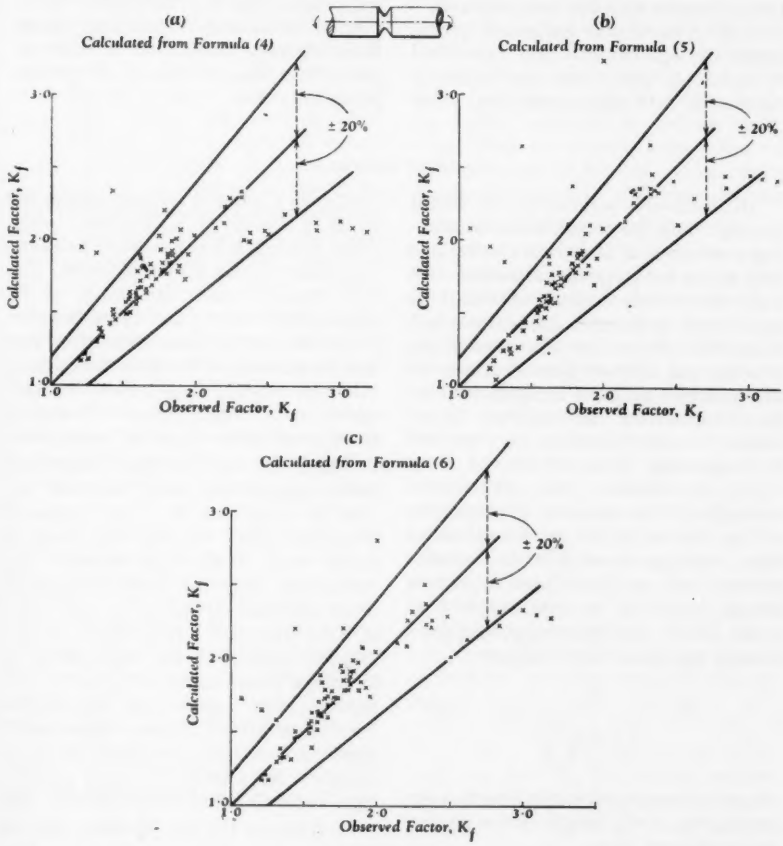

Figure 6. Comparison of observed and calculated strength reduction factors for grooved bars of various steels tested in fatigue.

are present. Although the difficulty is revealed by examining this hypothetical notch of exceedingly small size, the effect nevertheless persists for larger notches in the range of practical sizes. The greatest discrepancy probably occurs when notch size is of the same order as that of inherent flaws, for then no reduction in strength due to the notch would be expected.

Thus Neuber's formula appears to give too high a value of the effective s.c.f. for notches of a size comparable with that of inherent flaws, and so should not be used in such cases. As the formula may also give low values on account of flank-angle correction,

it may happen that the two errors cancel out in particular instances, giving apparent agreement. The modified formula (4) eliminates the difficulty associated with flank angle, and gives reasonable trends in size effect for all notches including V-notches, except those of very small size. It thus appears that this formula is of greater practical value.

PROPOSED FORMULAE

An alternative solution can be found by regarding the material as containing a number of stress concentrations due to its heterogeneous nature. Not only the crystals themselves with their anisotropic properties, dislocations and boundary effects, but also inclusions, cavities and surface discontinuities in the material produce irregularities in the micro-stress distribution. These stress concentrations can be regarded as originating from equivalent flaws of characteristic size distributed throughout the material. An analysis of the interacting effect of equivalent flaws with geometrical stress concentrations that are introduced leads to a simple empirical material notch formula, which has been suggested previously by the author,[6] namely

$$K_f = \frac{K}{1 + 2\sqrt{\frac{a}{R}}} \quad \ldots\ldots (5)$$

where a = material notch length, corresponding to the length of the equivalent inherent flaw.

This formula satisfies the limiting conditions, and although based on different concepts from Neuber's, it possesses similar characteristics as to size effect. Incidentally, by following the analysis and assumptions used by Neuber in deriving his formula, one can equally well arrive at formula (5), if correction for flank angle is omitted, for the reasons discussed above. Formula (5) overcomes the difficulty associated with the small notch, as shown in Fig. 3. As size of notch is reduced, so a point is reached where the value of the factor K_t becomes unity, when the notch has the same weakening effect as the inherent flaws. For even smaller sizes the value of K_t drops below unity, signifying that the notch under consideration produces less weakening effect than that due to inherent flaws of larger size. The material as a whole should therefore have an effective factor of unit value.

When the value of the theoretical factor approaches unity, the base radius of notch must be large compared with the other dimensions, such as notch depth. With large values of the radius, the value of the effective factor must approach unity in formula (5). If, however, the notch depth is extremely small and the base radius is of typical proportions, values of the effective factor may be very slightly below unity (this happens when notch depth is less than the material notch length.) As effective factors greater than unity should be obtained for this case, formula (5) can be elaborated to

$$K_f = \frac{K}{1 + 2\sqrt{\frac{a}{R}\left(\frac{K-1}{K}\right)}} \quad \ldots (6)$$

This ensures that when the theoretical factor has the value of unity, the effective factor must also have the value of unity, and thus the formula gives realistic trends for notches of extremely small depth.

A comparison between the three formulae (4), (5) and (6) is made in Fig. 4 for a series of geometrically-

similar specimens of different size. For Neuber's simplified formula (4), the value of the material constant A is assumed to be 0.01 in., this being appropriate to fatigue tests on a mild steel with strong influence of size. With formula (5) a material notch length of $a = 0.00085$ in. is assumed, and with formula (6), $a = 0.0019$ in. The comparison shows that general trends are very similar for all three formulae over most of the range of size.

In applying Neuber's formula to fatigue results, Kuhn and Hardrath have demonstrated that the value of the material constant A is approximately dependent upon the tensile strength of the steels used. The values that were suggested are given in Fig. 5, together with corresponding values of the material notch length for formula (5) and also for formula (6). By applying these values to a series of circumferentially grooved specimens tested in bending that has been given by Kuhn and Hardrath, a comparison between observed and calculated effective factors may be obtained, as shown in Fig. 6. The graphs demonstrate that all three formulae give the same order of accuracy, with the calculated value usually being within 20 per cent of the observed value for the very wide range of tests covered. Discrepancies could be due to the fact that notch sensitivity is not dependent simply on tensile strength, but it is more probable that the very high standards required in testing technique have not always been achieved in this series of results, which were obtained from many different sources.

By considering in similar manner the data provided by Kuhn and Hardrath for other types of notches, such as shouldered shafts and bars with transverse holes, it is found that all three formulae give approximately the same agreement with experiment, with formula (6) being very slightly better than formula (5), and this in turn is very slightly better than formula (4). The average discrepancy is of the order of 10 per cent. For design purposes it is considered that all three formulae are capable of assessing size effect to a degree of accuracy far greater than is likely to be required in practice, and so any one formula could be recommended. While there are some grounds for favoring the use of formula (6), there is a need for accurate fatigue data covering a wide variety of different types and sizes of notch for a few representative materials, so that the limitations and accuracy of formulae may be assessed.

CONCLUSIONS

Theoretical stress concentration factors for notched parts may be applied to practical strength values for fatigue loading, and for static loading of brittle parts. The application to the static loading of ductile parts is confined to estimations of local yielding in notches. The following three factors may be involved in the application.

(1) A combined stress criterion of failure applied to the stress conditions existing at the point of maximum stress. This produces a modified stress concentration factor which may be lower in value than the theoretical factor.

(2) A redistribution of stress due to inelastic behavior, which may occur even under fatigue conditions. The magnitude of the effect depends on the material and condition of test.

(3) A size effect in notched parts, which is associated with the stress gradients at the point of maximum

stress. The magnitude of the effect depends on the size of notch, the material and the conditions of test.

For most design problems involving the determination of the fatigue limit of notched parts it is sufficient to allow only for size effect as given in item (3) above, when behavior can be predicted by use of a simple formula involving a characteristic notch sensitivity constant.

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MACHINERY FOR CARGO LINERS

ACKNOWLEDGEMENT

This is an abridged version of the 27th Thomas Lowe Gray lecture read before the Institution of Mechanical Engineers on 21 January 1955. The author is CDR (E) L. Baker, R.N. (Ret'd), Chief Superintendent Engineer, Blue Funnel Line. The article appeared in the February 1955 issue of *The British Motor Ship*.

Rather more than 20 years ago Cripps (1930) presented a paper on the broad economic factors of the operation of cargo liners and this was followed 18 months later by some further consideration of the problem by Davis (1932). Much of what was then written is still true and this lecture is largely concerned with revising the data then presented.

SELECTION OF MACHINERY FOR THE DESIGN

It will be appreciated that a ship is a compromise—how much of a technical compromise it is may be clearer later, when an example is worked out. In the meantime, the superintendent engineer has to provide the naval architect with his requirements for weight and space of machinery and fuel consumption.

The established practice of different companies varies widely, largely owing to the differing experiences and the different financial policies. For example, a company such as the Blue Funnel Line operates the fleet on an uninsured basis and, consequently, considerable care is exercised in all phases of the design, construction, and operation of the fleet: forward thinking and development are encouraged when the potential return in the form of lower fuel cost, or reduced weight and space appear to justify the risk to be taken. Directly as a consequence of these policies come ships that outlast their contemporaries of more conventional design.

The most important contribution to the daily operating cost will be that of fuel, with second order variables for capital and maintenance costs.

Gas turbines must be ruled out at present because they must have a distillate fuel for long-life operation.

The average cost of marine Diesel fuel is about half as much again as boiler fuel, that is, a fuel rate of 0.4 lb. per b.h.p. per hr. on marine Diesel fuel costs as much per day as 0.6 lb. per b.h.p. per hr. on bunker C. It follows, therefore, that a Diesel installation utilizing marine Diesel fuel can only compete with steam turbines when the aggregate weight of the machinery, water, and fuel for the selected voyage and the space occupied by the machinery are equal to or less than those for the steam plant. The power at which this occurs is rather less than 7,000 s.h.p. with direct-drive machinery, and with geared-drive, using Diesel engines rotating at less than about 750 r.p.m. Above this power the Diesel engine must burn boiler fuel and in doing so has once more achieved a gain in economy over the steamship.

For cargo-liner propulsion, Diesel-

electric and turbo-electric drives cannot compete with the mechanical-coupling system simply because the capital cost must be greater, the fuel consumption as great or greater, and the weight and space certainly not less and usually greater.

The high-speed geared Diesel has not hitherto found favor in the power ranges required for liner propulsion because the engines at present developed and established have insufficient reliability; this necessitates excessive maintenance which in turn throws an undue burden on operating personnel. The advent of the Deltic engine, which is small enough to envisage a technique of the replacement of whole units, with the minimum of shipboard maintenance, may well justify the position being reviewed in the future.

The present remaining protagonists are, therefore, the direct-drive Diesel engine and the steam turbine.

The fuel consumption in port is between 7 and 12 tons a day for steam winches compared with 1½-2 tons a day for electric winches. A difference of 5 tons a day for half the year is equivalent to an extra fuel cost of £100,000 over the life of the ship and it is clearly impossible to justify such an amount in any but the most unusual cases.

PROGRESS OF DIESEL MACHINERY

The progress of the Diesel engine for marine propulsion has been quite astounding. In roughly 30 years the power installed in a given engine room has more than trebled as can be seen from Table 1, which applies to ships of the same company and the same engine designer.

Table 1.—Progress of Diesel Propulsion in the Blue Funnel Fleet.

Date of construction	Number of screws	S.h.p. per shaft	Length of engine room, feet
1922	2	2,400	63
1926	2	3,300	60
1929	2	4,300	54
1938	2	6,000	66
1945	1	6,800	57
1949	1	7,200	57
1955	1	8,000	51

There has been steady development in the power taken together with a number of clearly defined changes of a more radical nature. In 1929 the four-cycle engine was supercharged; in 1933 the two-cycle engine was introduced in the double-acting form and it is still the most compact and the lightest. In 1945 the "coverless" engine was built and, in 1949, the two-cycle single-acting engine was adopted to gain some simplification of maintenance at the expense of weight. The most recent designs which are due to enter service in 1955 are exhaust-turbine supercharged versions of the 1949 design.

One feature of the particular company's policy since the 1939-45 war has been the adoption of an increasing measure of standardization even, in a few instances, at the expense of a slight loss in efficiency or of extra weight.

For example, since 1949 the main engines for all motor ships to be delivered and ordered have been of the same bore and stroke; the Diesel generators for motor ships were of the same type from 1947 until 1954 when a new standard was adopted based on that fitted to the three 950° F. steamships of the "Nestor" class.

The effect on the maintenance cost of supercharging a two-cycle engine has not yet been determined but experience with the four-cycle supercharged engines gives hope that a considerable increase in rating may be obtained without deleterious effects. In any event, the blower power will be re-

moved from the crankshaft with consequent effect on the net power for the same rating.

Finally, progress has been made in the weight and thermal efficiency of Diesel engines as indicated in Table 2.

Table 2.—Weight and Thermal Efficiency of Diesel Machinery.

Year of completion	Machinery	Motor ships maximum continuous shaft power, s.h.p.	Machinery weight, lb. per b.h.p.	Fuel rate, lb. per s.h.p. for all purposes	Efficiency, per cent
1929	T.S. Diesel SA4S supercharged	8,000	450	0.40	34.3
1936	S.S. Diesel DA2S	3,100	408	0.42*	32.7
1938	T.S. Diesel DA2S	11,800	364	0.40	34.3
1943	S.S. Diesel DA2S	8,000	351		
1945	S.S. Diesel DA2S	6,800	363	0.39	35.2
1947	S.S. Diesel DA2S	6,800	370		
1950	S.S. Diesel SA2S	7,000	439	0.385	35.7
1955	Sketch design SA2S supercharged	8,000	327	0.38 (?)	36.1 (?)

CYLINDER-LINER WEAR

The crux of the economics of burning boiler oil in Diesel engines lies in the cylinder-liner wear. A simple, four-stroke engine shows the same figures for liner wear, irrespective of the fuel burnt, but the double-acting two-stroke engine has a greatly increased wear rate on boiler oil despite the use of detergent cylinder oils.

An endeavor was made to establish the effective rate of wear, but the variable factors appeared to be such that no satisfactory conclusion could be reached. The first step to eliminate the sources of error was to appoint one man to take all the wear measurements since it was found that the errors of measurement as between individuals outweighed the wear due to 1,000 hours of operating life. Since that date, a limited number of liners had been worn to the limits and it had become apparent that the next difficulty was that although 160 thousandths of an inch total wear was allowed, the availability required of the machinery meant that few if any of the liners were removed at a wear of exactly 160 thousandths.

PROGRESS OF STEAM MACHINERY

As the Blue Funnel Line had built no steamships for 25 years, the progress to be recorded is of shorter duration. There are three classes of ship involved, all of postwar construction as shown in Table 3.

One of the most unsatisfactory features of two of the three classes of steamship is the wide range of fuel consumptions found in service. The Admirality coefficients are not as widely spread as those of Diesel ships but, whereas the specific fuel rate of Diesel ships varies about 3%, that of the two classes of steamship varies as much as 10% from ship to ship. In the third case the variation appears to be very much less.

* Increase due to disproportionate hotel load in this class of ship.

Table 3. PROGRESS OF MACHINERY IN STEAMSHIPS

Year of completion	Steam conditions	Maximum continuous power, b.h.p.	Length of engine room, feet	Machinery weight, lb. per s.h.p.	Fuel rate, lb. per s.h.p. for all purposes	Thermal efficiency, per cent
1949	S.S. turbine 450/750 deg. F.	6,800	58	250	0.61-0.65	22.5-21.1*
1949	S.S. turbine 525/850 deg. F.	15,000	63	209	0.56-0.61	24.5-22.5*
1952	S.S. turbine 625/950 deg. F.	8,000	51	243	0.53	25.9
—	Sketch design S.S. turbine 625/950 deg. F.	11,500	54	207	0.52(?)	26.4(?)

It is, of course, essential to avoid stultification of progress, but if changes are made with intelligence, a considerable measure of interchangeability can be achieved without preventing worthwhile development. A brief history of the "A" class serves to illustrate the point.

As originally conceived, the "A" class of motor ship was propelled by an eight-cylinder double-acting two-cycle Diesel engine of 6,800 b.h.p. The generators were three five-cylinder engines of 220 kW. output each, and the lubricating-oil pumps were two of 350 tons per hr. output.

After the first 12, the main engine was changed to a seven-cylinder single-acting engine of larger bore and stroke, giving 7,000 b.h.p., but the other units remained the same. After the first four of the modified "A" class, three 250 tons per hr. lubricating-oil pumps were substituted for the two pumps in order to enable a complete separation of the bearing and piston cooling systems to be achieved. After the ninth of the modified "A" class, the main engine is again being changed so that 8,000 b.h.p. can be achieved as a supercharged six-cylinder version of the seven-cylinder engine, the liners and pistons being interchangeable. The tailshaft and propeller of the latest design are interchangeable with the seven-cylinder ships and with the "Nestor" and "Nelus," the two steamers of 8,000 s.h.p., but not with the original ships.

At the same time as this change is being made, the policy in regard to the generators has been changed so that two 450-kW. Vee 8 generators replace the three 220-kW. generators. In all essential parts, except the blowers, these new engines are identical with the 350-kW. engines fitted in the steamships.

Of the items of costs within the control of the technical staff, it is of interest to compare the proportions of the major items (Table 4).

Table 4. BREAKDOWN OF COSTS OF A VOYAGE (1953)

Item	'P' and 'H' class steam turbines, per cent	'A' class motor ship, per cent
Fuel at full speed	26.7	17.0
Fuel in port	4.2	2.6
Wages	11.5	15.9
Victualling	3.3	4.7
Repairs	9.8	11.6
Stores	2.9	4.1
Cargo expenses	41.6	44.1
	100	100

*Range of performance covers number of nominally identical ships and machinery.

EFFECTIVE COST OF FUEL

Reference has previously been made to the price value of marine Diesel and bunker C fuels in New York, but this is, of course, only true if bunkering is taking place in New York. With world-wide trade, a number of important factors enter; the shipowner naturally tries to bunker as much fuel as possible at the cheapest ports but he is frequently frustrated in this endeavor by considerations of the available deadweight.

The effective prices of boiler and Diesel fuel and their ratio are shown in Table 5.

Table 5. EFFECTIVE PRICES OF FUEL

	1950				1951				1952				1954
	First quarter	Second quarter	Third quarter	Fourth quarter	First quarter	Second quarter	Third quarter	Fourth quarter	First quarter	Second quarter	Third quarter	Fourth quarter	Second quarter
	s. d.	*s. d.*	*s. d.*	*s. d.*	*s. d.*	*s. d.*	*s. d.*	*s. d.*	*s. d.*	*s. d.*	*s. d.*	*s. d.*	*s. d.*
(A)	164 6	151 3	160 0	165 5	183 0	182 2	193 10	198 10	205 5	217 10	217 5	205 10	201 0
(B)	124 2	122 4	122 0	125 0	120 10	126 5	134 2	142 2	145 5	150 0	151 0	138 7	124 0
(C)	1.32	1.285	1.31	1.32	1.515	1.44	1.44	1.500	1.41	1.45	1.44	1.485	1.62

(A)—Price per ton of Diesel fuel (world mean)
(B)—Price per ton of boiler fuel (world mean)
(C)—Ratio of prices of Diesel and boiler fuel

EVOLUTION OF A DESIGN

From the broad statement of the problem and the accepted limitations, the naval architect proceeds on the basis of past designs to "cut and try." At the same time, the superintendent engineer is preparing alternative machinery arrangements based on a preliminary estimate of the power and using any or all of the possible combinations of types of propulsion.

When the naval architect and the superintendent engineer have completed their development of the designs, the owners have to select the most economic alternative for final development into completed basic plans and specification.

A hypothetical case for a vessel with a minimum service speed of 16.5 knots, a bale capacity of 14,500 tons at 40 cu. ft. minimum and a deadweight of 8,500 tons on winter draught weather is set out in Table 6.

Table 6. COMPARSIONS OF DESIGN WITH REQUIREMENTS

a

Item	Steam-turbine ship	Diesel motor ship	Existing ship
Hull weight, tons	5,855	5,905	4,518
Machinery, tons	960	1,510	1,262
Deadweight, tons	10,285	9,685	9,295
Summer displacement tons	17,100	17,100	15,075

b

Item	Steam-turbine ship	Diesel motor ship	Existing ship Design	Existing ship Actual
Speed, knots	17	17	15.5 plus	16.17
Power, s.h.p.	10,500	11,000*	5,720	6,720
Fuel cons. tons per day	58	47	27.9	27.8

* An allowance of 5 per cent extra power to ensure margin for effect of weather from data on previous ships.

EFFECT OF FUEL ON THE CHOICE OF MACHINERY

It will have become apparent from the above that in the fringe zone the type and quality of the fuel burnt in the propulsion plant exerts a controlling influence on the type of machinery to be fitted.

Basically, however, this is no new problem to the shipping industry, which became so accustomed to the adjustment of prices between coal and oil as to keep the field open for exploitation of both fuels. The end was, of course, eventually attained when coal was priced out of reach after the 1939-45 war. Possibly, a decision in this new case will eventually be reached in the same way, for the price ratio of Diesel oil to boiler oil at Aden has now reached 1.62, and differential prices for various grades of boiler fuel may well spread to the more popular bunkering ports.

The author has tried, within the scope of a short lecture, to indicate some of the factors that enter into the selection of machinery for ships. It was difficult to do more than sketch in the general outline.

Table 7. Economic Balance for 25,000-Mile Voyage

Item	Steam-turbine ship, £	Diesel motor ship, £
Fuel, 687 tons costing 124s. 0d. per ton	4,259	—
Lubricating oil (differential)	—	855
Maintenance (differential)	—	2,800
Canal dues (differential)	225	—
Totals	4,484	3,655
Net extra cost of turbine per voyage	3,655 829	
Extra earning capacity	350 tons bale, 425 tons dead-weight	

U ANTENNA TRANSMISSION-LINE

COMMANDER CHARLES W. HARRISON, JR., U.S.N.

(Authors biography on page 579)

SUMMARY: The scope of the method for analyzing folded antennas presented in a recent paper is enlarged to include structures possessing no planes of symmetry. This is accomplished by separating the current and vector potential existing along each conductor into even and odd parts. A formula is developed for the driving point admittance of a U antenna transmission-line to illustrate the procedure.

The admittance of a folded antenna that is symmetrical with respect to the driving point may be determined by a systematic application of the procedures set forth in an earlier paper.[1] When the structure is asymmetrical it is desirable to utilize a somewhat different approach to the solution of the problem. This is not to imply that the method presented in the basic article is abandoned, but rather extended to encompass a very large variety of folded wire configurations that cannot be analyzed conveniently by the techniques previously advanced. This extension consists in adding another step; that of separating the current and vector potential along each conductor into even and odd parts.

To illustrate the principle an antenna transmission-line possessing no planes of symmetry has been selected. This structure consists of two isolated cylindrical conductors of radius **a** and half-length **h** spaced a distance **b** apart as shown in Figure 1. The wires are connected together at one end. One conductor is center-driven. The inequalities $a<<h$, $\beta a<<1$ and $\beta b<<1$ obtain. β is 2π divided by the wavelength λ. The configuration chosen presents the simplest problem for which the solution is facilitated by employment of the even and odd current and vector potential approach. This achieves brevity in presentation. Furthermore, the validity of the final result may be checked easily by the interested reader, utilizing the principle of superposition of circuits.

The preliminary steps in the analysis, as well as details concerning a transformation of the integral equations are omitted, it being presumed that the reader has studied reference 1. The notation employed in both papers is made identical insofar as practicable so that common symbolism need not be redefined.

The electrical properties of the U antenna transmission-line may be deduced from the simultaneous integral equations for the antenna currents:

[1] Charles W. Harrison, Jr., "Driving Point Admittance of a Symmetrically Folded Antenna," Journal of the American Society of Naval Engineers, Vol. 67, No. 1, pp. 213-238, February, 1955.

$$A_1(z) \propto \int_{-h}^{h} I_1(z') K_a(z, z')\,dz + \int_{-h}^{h} I_2(z') K_b(z, z')\,dz' = -j\frac{4\pi}{R_c}\left\{C_1 \cos \beta z + C_2 \sin \beta z + \frac{V_o}{2}\sin \beta/z/\right\} \quad (1)$$

$$A_2(z) \propto \int_{-h}^{h} I_1(z') K_b(z, z')\,dz' + \int_{-h}^{h} I_2(z') K_a(z, z')\,dz' = -j\frac{4\pi}{R_c}\left\{C_3 \cos \beta z + C_4 \sin \beta z\right\} \quad (2)$$

Here

$$K_a(z, z') = \frac{e^{-j\beta\sqrt{(z-z')^2+a^2}}}{\sqrt{(z-z')^2+a^2}} \quad (3)$$

Equations (1) and (2) apply to wires (1) and (2), respectively. They are obtained by equating the Helmholtz integral for the total vector potential at a point on the surface of a wire to the solution of the differential equation for the total vector potential at the same point.

The vector potentials $A_1(z)$ and $A_2(z)$ may be written

$$\left.\begin{aligned} A_1(z) &= \frac{1}{2}\left\{A_1(z)+A_1(-z)\right\} + \frac{1}{2}\left\{A_1(z)-A_1(-z)\right\} = A_{1\,\text{even}}(z)+A_{1\,\text{odd}}(z) \\ A_2(z) &= \frac{1}{2}\left\{A_2(z)+A_2(-z)\right\} + \frac{1}{2}\left\{A_2(z)-A_2(-z)\right\} = A_{2\,\text{even}}(z)+A_{2\,\text{odd}}(z) \end{aligned}\right\} \quad (4)$$

The currents $I_1(z)$ and $I_2(z)$ may be separated into even and odd parts in the same manner. Thus

$$\left.\begin{aligned} I_1(z) &= I_{1\,\text{even}}(z)+I_{1\,\text{odd}}(z) \\ I_2(z) &= I_{2\,\text{even}}(z)+I_{2\,\text{odd}}(z) \end{aligned}\right\} \quad (5)$$

On separating equations (1) and (2) into even and odd parts (using (4) and (5)), and employing the transformation technique explained previously (reference 1), one obtains

$$J_{d\,\text{even}}(z)+I_{1\,\text{even}}(z)\lambda_a+I_{2\,\text{even}}(z)\lambda_b = -j\frac{4\pi}{R_c}\left\{C_1 \cos \beta z + \frac{V_o}{2}\sin \beta/z/\right\} \quad (6)$$

$$J_{d\,\text{even}}(z)+I_{1\,\text{even}}(z)\lambda_b+I_{2\,\text{even}}(z)\lambda_a = -j\frac{4\pi}{R_c}C_3 \cos \beta z \quad (7)$$

$$J_{d\,\text{odd}}(z)+I_{1\,\text{odd}}(z)\lambda_a+I_{2\,\text{odd}}(z)\lambda_b = -j\frac{4\pi}{R_c}C_2 \sin \beta z \quad (8)$$

$$J_{d\ odd}(z)+I_{1\ odd}(z)\lambda_b+I_{2\ odd}(z)\lambda_a=-j\frac{4\pi}{R_c}C_4\sin\beta z \quad \ldots\ldots (9)$$

Here

$$J_{d\ even}(z)=\int_{-h}^{h} I_{T\ even}(z')K_d(z,z')\,dz' \quad \ldots\ldots (10)$$

$$\text{where } I_{T\ even}(z)=I_{1\ even}(z)+I_{2\ even}(z) \quad \ldots\ldots (11)$$

and

$$J_{d\ odd}(z)=\int_{-h}^{h} I_{T\ odd}(z')K_d(z,z')\,dz' \quad \ldots\ldots (12)$$

where

$$I_{T\ odd}(z)=I_{1\ odd}(z)+I_{2\ odd}(z) \quad \ldots\ldots (13)$$

In (6) to (9) inclusive,

$$\left.\begin{aligned}\lambda_a&=2\log\frac{d}{a}\\ \lambda_b&=2\log\frac{d}{b}\end{aligned}\right\} \quad \ldots\ldots (14)$$

Adding (6) and (7) and setting $\lambda_a=-\lambda_b$ so that

$$d=\sqrt{ab}<b \quad \ldots\ldots (15)$$

$$J_{d\ even}(z)=-j\frac{4\pi}{R_c}\left\{\frac{(C_1+C_3)}{2}\cos\beta z+\frac{V_o}{4}\sin\beta/z/\right\} \quad \ldots\ldots (16)$$

This equation is in King-Middleton[2] form provided $I_{1\ even}(h)+I_{2\ even}(h)=0$. This matter will be investigated later. Adding (8) and (9) and using (15),

$$J_{d\ odd}(z)=-j\frac{4\pi}{R_c}\left\{\frac{C_2+C_4}{2}\right\}\sin\beta z \quad \ldots\ldots (17)$$

The driving point admittance of the U antenna transmission-line is the ratio of $I_{1\ even}(0)$ and V_o. This is so because $I_{1\ odd}(0)=0$. ($I_1(0)=I_{1\ even}(0)+I_{1\ odd}(0)$.) Hence the solution of (17) is not required in the present problem. If the current distribution along the wires is to be studied, then the solution of (17) is mandatory.

Subtracting (7) from (6) and then (9) from (8) leads to the transmission-line expressions

$$I_{1\ even}(z)-I_{2\ even}(z)=-j\frac{4\pi}{\lambda_{ab}R_c}\left\{(C_1-C_3)\cos\beta z+\frac{V_o}{2}\sin\beta/z/\right\} \quad \ldots\ldots (18)$$

and

$$I_{1\ odd}(z)-I_{2\ odd}(z)=-j\frac{4\pi}{\lambda_{ab}R_c}\left\{(C_2-C_4)\sin\beta z\right\} \quad \ldots\ldots (19)$$

[2] R. King and D. Middleton, "Cylindrical Antenna; Current and Impedance," Quarterly of Applied Mathematics, Vol. III, No. 4, pp. 302-335, January 1946.

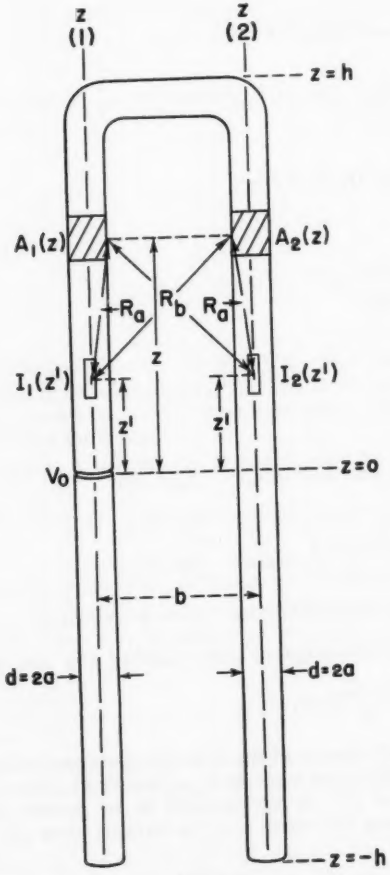


Figure 1
U Antenna Transmission-Line

Here

$$\lambda_{ab}=\lambda_a-\lambda_b=2\log\frac{b}{a} \quad \ldots\ldots (20)$$

Continuity of charge in the shorting bar requires

$$\frac{\partial}{\partial_z}\left\{I_{1\ even}(z)+I_{1\ odd}(z)-I_{2\ even}(z)-I_{2\ odd}(z)\right\}_{z=h}=0 \quad \ldots\ldots (21)$$

Application of (21) leads to one equation relating the constants. It is

$$-(C_1-C_3)\sin\beta h+\frac{V_o}{2}\cos\beta h+(C_2-C_4)\cos\beta h=0 \quad \ldots\ldots (22)$$

An inspection of the Figure 1 shows that

$$I_{1\ even}(h)+I_{1\ odd}(h)+I_{2\ even}(h)+I_{2\ odd}(h)=0 \quad \ldots\ldots (23)$$

and

$$I_{1\ even}(-h)+I_{1\ odd}(-h)+I_{2\ even}(-h)+I_{2\ odd}(-h)=0 \quad \ldots\ldots (24)$$

Now from symmetry considerations

$$\left.\begin{aligned} I_{1\ even}(h)&=I_{1\ even}(-h)\\ I_{1\ odd}(h)&=I_{1\ odd}(-h)\\ I_{2\ even}(h)&=I_{2\ even}(-h)\\ I_{2\ odd}(h)&=-I_{2\ odd}(-h)\end{aligned}\right\} \quad \ldots\ldots (25)$$

Using (25), (24) becomes

$$I_{1\ even}(h)-I_{1\ odd}(h)+I_{2\ even}(h)-I_{2\ odd}(h)=0 \quad \ldots\ldots (26)$$

Equations (23) and (26) require

$$I_{1\ even}(h)+I_{2\ even}(h)=0 \quad \ldots\ldots (27)$$

and

$$I_{1\ odd}(h)+I_{2\ odd}(h)=0 \quad \ldots\ldots (28)$$

Expression (27) proves that (16) is in true King-Middleton form. The solution of the latter equation is

$$Z_d=\frac{V_K}{I_{T\ even}(0)}=\frac{V_o}{2I_{T\ even}(0)} \quad \ldots\ldots (29)$$

Setting z=h in (18) and (19) yields

$$I_{1\ even}(h)-I_{2\ even}(h)=-j\frac{4\pi}{\lambda_{ab}R_c}\left\{(C_1-C_3)\cos\beta h+\frac{V_o}{2}\sin\beta h\right\} \quad \ldots\ldots (30)$$

and

$$I_{1\ odd}(h)-I_{2\ odd}(h)=-j\frac{4\pi}{\lambda_{ab}R_c}\left\{(C_2-C_4)\sin\beta h\right\} \quad \ldots\ldots (31)$$

Subtracting (31) from (30),

$$I_{1\,even}(h)-I_{1\,odd}(h)-I_{2\,even}(h)+I_{2\,odd}(h)=-j\frac{4\pi}{\lambda_{ab}R_c}\left\{(C_1-C_3)\cos\beta h + \frac{V_o}{2}\sin\beta h-(C_2-C_4)\sin\beta h\right\} \quad (32)$$

Using (25) and the relations $I_{1\,even}(-h) + I_{1\,odd}(-h) = 0$ and $I_{2\,even}(-h) + I_{2\,odd}(-h) = 0$, it is easy to show that the left-hand side of (32) is zero. Hence the right-hand side is zero also. Thus

$$(C_1-C_3)\cos\beta h+\frac{V_o}{2}\sin\beta h-(C_2-C_4)\sin\beta h=0 \quad (33)$$

The simultaneous solution of (22) and (33) for (C_1-C_3) yields

$$C_1-C_3=-\frac{V_o}{2}\tan 2\beta h \quad (34)$$

Setting $z=0$ in (18), and rearranging,

$$I_{1\,even}(0)=I_{2\,even}(0)-j\frac{4\pi}{\lambda_{ab}R_c}(C_1-C_3) \quad (35)$$

Adding $I_{1\,even}(0)$ to both sides of (35) and introducing the value of (C_1-C_3) from (34),

$$I_{1\,even}(0)=\frac{I_{T\,even}(0)}{2}+j\frac{\pi}{\lambda_{ab}R_c}V_o\tan 2\beta h \quad (36)$$

$I_{T\,even}(0)$ is eliminated from (36) by (29). The final formula for the input admittance of the U antenna transmission line is

$$Y_{in}=\frac{I_{1\,even}(0)}{V_o}=\frac{1}{4Z_d}+j\frac{\tan 2\beta h}{240\log\frac{b}{a}}\text{ mhos} \quad (37)$$

Here Z_d is the input impedance in ohms of an isolated symmetrical center-driven antenna of half-length h and radius $d=\sqrt{ab}$. Equation (37) is valid provided $a<<h, \beta a<<1$ and $\beta b<<1$.

8,000-B.H.P. TWO-STROKE DIESEL ENGINE WITH EXHAUST-TURBINE SUPERCHARGING

ACKNOWLEDGEMENT

This is a staff article which appeared in the 4 February 1955 issue of *Engineering*.

During the past few weeks at Belfast, test-bed trials have been run of a single-acting two-stroke Diesel engine in which supercharging is by turbo-blowers driven by the exhaust gases of the engine. Harland and Wolff, Limited, the builders, state that the continuous service power is increased by 30 to 35 per cent. compared with normal induction. Full advantage is taken of the exhaust pulse from each cylinder: the volume of the exhaust pipe between the cylinder and the turbine is carefully chosen in order to obtain the maximum pulse energy by using the gases as nearly as practicable at the pressure and temperature of discharge from the cylinder. In addition, the fall in exhaust-manifold pressure after each pulse allows adequate scavenge air to pass through the cylinder, thus removing the residual exhaust gases and reducing the thermal stresses by cooling the piston and combustion space.

The arrangement is shown in simplified form in Fig. 1, applied to the six-cylinder Harland and Wolff engine. Two turbo-blowers are mounted on the back of the engine, and each is driven independently by the exhaust from three adjacent cylinders. The air delivered from the two compressors, however, is fed to a common induction manifold, from which it is delivered to the cylinders. It will be seen that the chain-driven Roots displacement-type scavenge-air blowers, which have always been characteristic of Harland and Wolff two-stroke engine, have been discarded. The engine is, indeed, the first single-acting two-stroke engine with piston-controlled exhaust ports to be turbo pressure-charged without the assistance of mechanically-driven scavenge pumps.

For driving the turbo-blowers when the Diesel engine is running at too low a speed for the exhaust gases to provide sufficient power, an electric motor is fitted to each blower. The motor cuts in and out automatically, as described later. This automatic motor arrangement has been provided at the request of the owners of the engine, Alfred Holt, Limited, but the builders

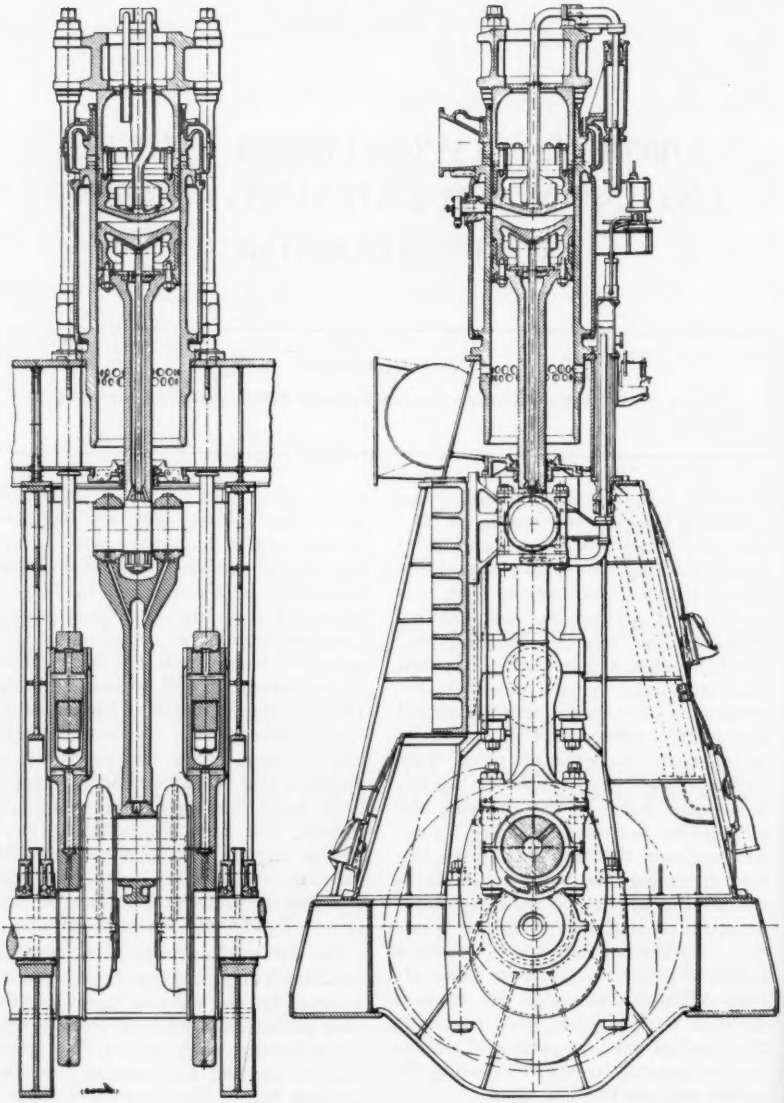


Figure 1. The general arrangement of the Harland and Wolff S.A.6 Diesel engine is similar, except for the supercharging, to their engine with normal air induction introduced five years ago.

would normally provide a small independently-driven air compressor for use under low-speed conditions.

A Serck cooler is fitted between each blower and the induction manifold (Fig. 2). This is necessary because the rise in temperature of the air as it passes through the compressor would otherwise prevent the increase in weight of charge from being proportional to the increase in pressure.

RANGE OF SIZES

The design of the new engine, apart from the turbo-charging arrangements, is generally similar to that of Harland and Wolff's normal-induction engine, which is a two-stroke crosshead-type single-acting opposed-piston engine. It has been extensively applied to many kinds of vessel, including large tankers, refrigerated cargo liners, cargo ships and tramps. It has also been installed, as a trunk engine, in cross-channel vessels and in power stations. As a propelling engine the two most popular cylinder sizes are: 750 mm. (29.53 in.) bore, 2,000 mm. (78.74 in.) combined stroke; and 620 mm. (24.41 in.) bore, 1,870 mm. (73.62 in.) combined stroke. With normal air induction, and with from four to eight cylinders, the range of power is thus 3,000 brake horse-power to 9,000 brake horse-power per screw, continuous sea ratings.

By pressure-charging, giving the 30 to 35 per cent increase in power, the same cylinder sizes will cover the

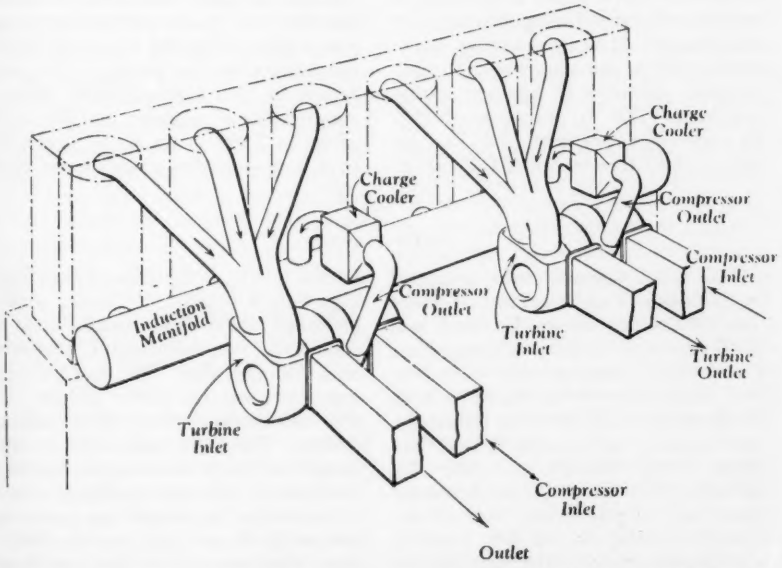


Figure 2. The turbo-blowers are so connected to the cylinders as to take full advantage of the separate exhaust pulses in charging and scavenging.

power range 4,000 brake horse-power to 14,000 brake horse-power from engines having four to nine cylinders. For the larger cylinder size, therefore, the output per cylinder is about 1,500 brake horse-power. This is equivalent to 9,000 brake horse-power for a six-cylinder unit of the type now on test, compared with 6,750 brake horse-power for a similar size engine with normal air induction. However, to comply with the owners' requirements for the first engine of the new type, the rating of this engine is very moderate; it will deliver 8,000 brake horse-power (British) or 8,100 brake horse-power (metric) at 112 r.p.m. The piston speed is 5.6 meters per second (1,100 ft. per minute), and the mean indicated pressure is 7 kg. per square centimeter (100 lb. per square inch). Other figures released so far are: fuel consumption 155 grams per brake horse-power per hour (0.34 lb. per brake horse-power per hour); firing pressure 52 kg. per square centimeter (740 lb. per square inch); compression pressure 41 kg. per square centimeter (580 lb. per square inch); air scavenge pressure 4.5 lb. per square inch; blower speed 5,500 r.p.m.

The engine has been designed to run continuously at sea on heavy fuel oils of viscosities up to 3,500 seconds (Redwood No. 1). Such fuels have been successfully burned for years in normal induction engine sizes with the Harland and Wolff positive cam-operated standard fuel-injection system. For the engine on test, however, Wilson and Kyle gas-compression fuel pumps and valves are fitted, in accordance with Blue Funnel Line requirements.

The weight of the six-cylinder engine, complete with turbo-blowers, over the crank ends, is 480 tons—a saving of 110 tons compared with a normal-induction engine. The turbo-charged engine is 40 ft. 8 in. over the frames, compared with 48 ft. 6 in. for the equivalent non-pressure-charged engine, a saving of 7 ft. 10 in.

It will be clear from the foregoing that the way is now open for twin-screw ships of 24,000 s.h.p. aggregate continuous service power to be propelled by two eight-cylinder direct-coupled Diesel engines, and for ships of up to 14,000 s.h.p. to be propelled by a nine-cylinder single-screw unit.

TURBO-BLOWERS

The turbo-blowers were supplied by D. Napier and Son, Limited, London, W.3, with whom Harland and Wolff have a license arrangement. The design is based on that of the Napier turbo-blowers which have been in use for several years on industrial, rail-traction and marine Diesel engines. These blowers are fully described, with drawings, performance data, etc., in publication No. 146 issued by D. Napier and Son, Limited. The blowers used on the Harland and Wolff engine are type MS600.

The MS600 turbo-blower, which is shown in Fig. 3, consists of four castings bolted together to house a single-stage axial-flow turbine and a centrifugal air compressor, mounted on a common shaft. The turbine castings have internal water jackets that are connected to the engine cooling system. The rotor assembly is supported in plain sleeve-type bearings mounted in spherical housings which permit some universal movement in sockets in the casings, and also facilitate maintenance of the bearings, which can be removed without disturbing the casings. The bearings are

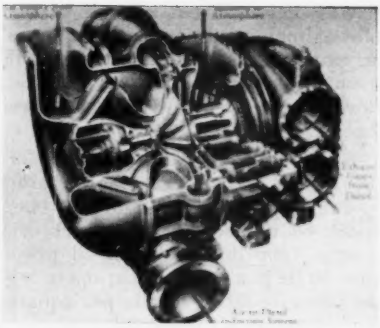


Figure 3. Showing the principle of the Napier MS 600 turbo-blower, which increases the power of the Harland and Wolff Diesel engine by 30 to 35 per cent.

pressure fed from an auxiliary-driven pump that delivers oil to connections on the end covers.

In the S.A.6 installation, the blowers are equipped with trunnion mountings in addition to the mounting brackets. These enable the blowers to be swung through 90 deg. from the normal position if required, so that during overhaul, components can be removed by a vertical lift. A combined air filter and silencer may be fitted in place of the compressor inlet casing; this provides ten filter panels that may be removed one at a time for cleaning while the turbo-blower is operating.

CONTROL MECHANISM

Under slow running conditions, when both the temperature and pressure of the exhaust gases from the engine are low, the energy released may be insufficient to maintain the turbo-blower output. This condition is aggravated during astern running when the air shortage due to the effect of the crank-angle phase difference on the astern port timing further diminishes the available exhaust energy. The blower is therefore fitted with a direct-current 10-h.p. motor that is automatically energized and engaged with the turbo-blower rotor shaft through an air-operated clutch below approximately 1,800 r.p.m. to ensure adequate output until the blower is self-supporting.

The blowers (and the blower driving motors when they are running) are pressure fed with lubricating oil from a self-contained system incorporating a dual-element filter and a cooler. The lubricating-oil pump and its driving motor are duplicated, and the two sets are so inter-connected through Igranic switches that if the one in service fails the other is automatically brought into operation.

When the engine-room comes on "stand-by" one of the lubricating-oil pumps is started and set to automatic control, the supply passing to the turbo-blower, but not to the driving motors until they are started through their respective starter boxes. As each blower motor is started, contacts in the starter box simultaneously actuate two solenoid valves; one opens the lubricating circuit to the motor bearings and the other admits air to the clutch, so engaging the motor with the turbo-blower rotor shaft and driving it until the Diesel engine is started and its exhaust gases are sufficient to produce a speed in excess of 1,800 r.p.m. As this speed is achieved, the tacho-generator fitted on the compressor end of the turbo-blower generates enough voltage to operate the relay, and this cuts off the power supply to the motor and closes the solenoid valve in the air line, so disengaging the clutch. At the same time the solenoid valve in the lubricating-

oil line shuts off the oil supply to the motor bearings without interrupting the supply to the turbo-blower bearings.

This condition prevails until the Diesel engine speed is reduced to the point at which the blower speed falls below 1,800 r.p.m., when the tacho-generator output fails to maintain the relay position and the motoring system is again brought into action.

The functioning of the system is monitored by signal lights which indicate that: (i) the blower motor(s) are energized; (ii) the Airflex clutch(es) are engaged; (iii) the lubricating-oil pump(s) are running; and (iv) the lubricating-oil pressure is satisfactory.

In addition to the automatic change-over switches interconnecting the duplicate lubricating-oil pumps, a flow switch is fitted in each of the blower/motor lubricating circuits as an extra safety precaution to shut off the appropriate blower motor if the oil flow is reduced below an acceptance level.

The principal dimensions, etc., of the Napier MS600 turbo-blower are: overall dimensions—length 92 in., width 66 in., height 70 in.; net dry weight 7,000 lb.; minimum oil pressure 40 lb. per square inch; operating air pressure 60 to 100 lb. per square inch; maximum turbine inlet temperature 1,200 deg. F. (650 deg. C.). Performance figures are given below.

	Maximum Continuous	Maximum One-Hour Limit
Speed, r.p.m.	6,400	7,360
Pressure ratio (nominal)	1.5:1	1.7:1
Sea-level delivery pressure (gauge), lb. per sq in.	7.35	10.3

TECHNICAL PROGRESS IN SHIPBUILDING DURING 1954

ACKNOWLEDGEMENT

This staff article was published in the January 1955 issue of *The Shipbuilder and Marine-Engine Builder.*

Before dealing in detail with the technical progress in shipbuilding during 1954, it is, perhaps, of interest to consider the output of this country against the background of world shipbuilding, and, in this connection, some useful information can be obtained from the latest data available at the time of going to press, *viz.*, the figures issued by Lloyd's Register of Shipping for the tonnage under construction at the end of September, 1954. The world totals at that time consisted of some 1,065 merchant ships, of 5,499,078 tons gross. Of this total, 316 ships of 2,058,457 tons, were under construction in Great Britain and Northern Ireland. These figures represented 37.43 per cent of the world total. When compared with the corresponding period for 1953, the world total shows a reduction of 500,000 tons gross, and while Britain's total was correspondingly smaller, her percentage of the world total was slightly greater than in 1953. It is very gratifying, therefore, to note that, in spite of high costs, Britain's position in world shipbuilding is being maintained.

According to Lloyd's returns, Germany occupies second place in the world shipbuilding picture, with 154 ships, representing 12.16 per cent of the total, while the Netherlands are third, with 130 ships, representing 8.54 per cent of the total. Although Britain's nearest competitors are only building about one-third of the tonnage that she is, the challenge of cheaper ships should not be ignored, and every effort should still be made by all branches of the industry to keep down building costs, while, at the same time, maintaining the excellence of the product.

By far the largest percentage of the tonnage being built consists of oil-tank vessels, the world total being 258 ships of 3,218,760 tons gross. Britain's share in this total is 83 vessels, of 1,093,530 tons. These are divided almost equally, so far as numbers are concerned, between steamships and motorships, but, clearly, the larger vessels, on the average, are steam-driven, as the steam tonnage is 678,165, as compared with the motorship tonnage of 415,365. Cargo liners occupy second place in British shipbuilding, 72 vessels of nearly 500,000 tons being under construction. Passenger ships, and passenger and cargo liners, are next in the list, there being 18 ships of more than 206,000 tons gross in this class, under construction.

Several notable vessels have either been launched or delivered during 1954. Mention might be made of the Cunard liner *Saxonia,* which entered service on the Liverpool-Quebec and Montreal run in September last. This ship, which is over 600 ft. in length, carries 125 first-class and 800 tourist-class passengers.

Another vessel of interest is the Orient liner *Orsova,* with a length of 668 ft. This vessel has a service speed of 22½ knots, and carries 681 first-class and 813 tourist-class passengers.

Two other ships worthy of mention are the *Arcadia* and *Iberia,* both built for the P. & O. Line, London. Both of these ships have a length of 688 ft. and a speed of 22½ knots. They both carry approximately the same number of passengers, *viz.,* about 670 first-class and 730 tourist-class.

Finally, with regard to this type of ship, mention may be made of the twin-screw passenger steamship *Southern Cross,* which was launched by Her Majesty The Queen in August last. This ship, which is being built for the Shaw Savill & Albion Co., Ltd., London, has a length of 560 ft., and is intended to carry over 1,100 passengers at a service speed of 20 knots. Unique features of the design of the vessel are that there are no cargo holds, and that the single funnel is located aft.

The vessels mentioned here, by name, are indicative of the up-to-date nature of the designs being prepared by British shipbuilders, and in many of their features show the progress which has been made throughout the year under review.

Other progress of a much more specialized and technical nature has also continued, both in this country and throughout the world, during the past year. The main body of this work can be divided into two categories, *viz.,* structural strength, and resistance and propulsion.

STRUCTURAL STRENGTH AND WELDING

During 1954, the British Shipbuilding Research Association have pursued many research programs, and in the field of structural strength, the work at Glengarnock has played a large part.

The most recent contribution to the problem of ship structural members was made in a paper, entitled "Ship Structural Members—Part VI.," read by Mr. J. McCallum, B.Sc., before the North-East Coast Institution of Engineers and Shipbuilders, in October last. The tests described in this paper were of a more fundamental nature than some of the previous tests, and the object was to ascertain how far the simple beam theory applies to stiffened plating, and also to confirm the Glengarnock experimental technique. The sections used consisted of flat bars, welded to plating, and the majority of the specimens were freely supported at their ends. One specimen was of uniform cross-section, but the others had bracket-like appendages at the ends, which were welded to the stiffeners, but were unconnected to the end structure of the testing machine.

The general results of the tests indicate that the beam theory applies within close limits, provided that the stiffener and plating are of uniform cross-section. Where the cross-section changes, rapidly, however, as at brackets and at uniform sections in the vicinity of brackets, the simple beam theory does not hold good. In particular, that part of the bracket remote from the stiffener is very ineffective in contributing to stiffness. This is not surprising, since, theoretically, the deflection of a non-uniform beam depends upon the double integration of the curve M/I, and at the end of a

freely-supported beam, the bending moment, and hence M/I, are small. Therefore, the contribution of a bracket is likely to be small. It appears that, in so far as the stiffness of the beam, as a whole, is concerned, a strut of suitable cross-section, in association with an adequately strengthened stiffener in way of the strut, is approximately equivalent to a bracket, but no evidence has been provided as to the mimimum amount of stiffening required.

Corrugation of the bottom plating of ships is a phenomenon which is comparatively new, and is most noticeable in all-welded ships. Since the war, this problem has caused some trouble; and it is, therefore, of considerable interest to have the paper on this subject, entitled "Corrugation of Bottom Shell Plating," which was read by Mr. J. M. Murray, M.B.E., B.Sc., before the Institution of Naval Architects, in April last. The ships in which corrugations were first observed had dimensions of 416 ft. by 56.8 ft. by 38 ft. In service, they were subjected to considerable hogging stresses, the total stress in heavy weather, due to water pressure and longitudinal bending, being of the order of 6 tons per sq. in. These stresses, imposed upon plating which already had some initial deflection, produced the corrugations. The depths of the corrugations, which were evident over the half-length amidships, were of the order of 1½ in., and indentation was generally upwards between the floors.

To deal with the problem of repairing these ships, the bottom plating was stiffened so as to reduce the panel size, and, for this purpose, inverted angle stiffeners (6 in. by 3½ in. by ½ in.) were welded to the shell. These stiffeners, together with the side and center girders, formed panels about 4 ft. in breadth in the transverse direction.

In order to investigate the problem thoroughly, Lloyd's Register of Shipping carried out experimental work on sections of the double bottom of an all-welded ship, and a ship of riveted construction. A theoretical solution was also attempted, and there was considerable success in this connection, in that it was possible to obtain good agreement between experimental and theoretical results.

The general conclusions arrived at from this very able piece of work are that the transversely-framed all-welded ship is inherently liable to corrugation of the bottom plating, and that the same phenomenon, while observable in riveted ships, is much rarer. Further, the only structural means of eliminating this species of failure is to introduce longitudinal framing. Once initial deflection has been sustained, the deflections will go on increasing over a period of years; so that efficiency of the hull structure is materially reduced, resulting in stresses which are high, unless the ship is stiffened by longitudinal panel breakers.

The results of a unique and valuable series of structural model tests have been made available during the past year, in a paper entitled "Ship Model Tests to Determine Bending Moments in Waves," read by Mr. Edward V. Lewis, before the American Society of Naval Architects and Marine Engineers, in November last.

These tests consisted of preparing a special model of a T2-SE.A1 tanker, and running it in ahead and following seas. The model was so constructed that the bending moment could be measured. Experiments of this nature offer an opportunity of checking the bending moments obtained from the usual longitudinal-strength calculation, and are not open to the doubt which must exist in attempting to correlate calculations with stresses and bending moments obtained for a ship in actual sea waves.

The model, in the experiment, was

made of wood and was cut in halves at the midship section. These two portions were pivoted, and a cantilever arrangement was so fitted that the relative rotation of the two parts could be measured. This rotation is clearly proportional to the bending moment applied at the midship section. The ⅛-in. gap between the fore and after-bodies of the model was covered by a watertight rubber seal, to exclude water. The vertical accelerations of the model at the fore and after ends were also measured by means of accelerometers, while the wave profile, undisturbed by the presence of the model, was also measured.

A brief note of the general conclusions obtained from these ingenious experiments is fitting. It was found that, in regular swells of ship length and of height equal to one-twentieth of the length, the dynamic bending moment amidships may be expected to be appreciably less than the values obtained by the ordinary static calculation, even when the Smith correction has been made. At speeds in the vicinity of synchronous heaving and pitching motion, a further reduction in the bending moment is to be expected, and, at the same time, there will be a tendency for slamming to occur.

In irregular storm seas having an average wave length somewhat greater, and an average height somewhat less, than the standard L/20 wave, peak bending moments greater than those in regular swells are to be expected. A point to be noted, however, is that the bending moments in irregular waves were never greater than those for the "standard" calculation, except when the Smith correction was made to the latter. It appears that bending moments reduced with the increasing violence of the model motions and it would seem, therefore, that the dynamic effects are favorable from a strength point of view, but only at the expense of increased slamming. The effect on the bending moment of ahead speed is small with following seas, but considerable when the waves are meeting the ship. There is a distinct upward trend of bending moment as the speed in ahead seas is increased beyond the service speed (in this case, 14½ knots).

We have given the results of these experiments in some detail, because of the obvious importance of the conclusions, and because the method described in the paper seems to open up a very considerable field for research in the longitudinal strength of ships. A study of the technique of these experiments is certainly worth while, and no doubt the future will see more work of this nature being carried out.

Details of another strength investigation, this time on full scale, have been published during the past year, in a paper entitled "Stress Investigation on a Tanker during Launching," read by Mr. A. J. Johnson, B.Sc., A.C.G.I., and Mr. M. Meek, B.Sc., before the North-East Coast Institution of Engineers and Shipbuilders, in January last. This was a British Shipbuilding Research Association's investigation, and concerned the stresses in a tanker of 11,000 tons deadweight during launching. The main objects of the investigation were to compare calculated stresses in the hull of the ship with actual stress observations made at the launch.

The ship was calibrated by carrying out a simple static test upon it, after being launched. In this way, it was possible to determine the strain corresponding to any particular bending moment. Strain was measured during the launch by means of 100-in. base strain gauges, giving an autographic record. A comparison of the calculated stresses, and those derived from the observed strain, showed, in general, that the character of the curves was the same, but the magnitudes were different. For example, the stress at

the instant when the stern lifted, according to theory, was of the order of 1.9 tons per sq. in. in the deck, while the calculated stress was only 1.5 tons per sq. in. Similarly, the stress in the bottom shell at this instant was about 1.6 tons per sq. in., whereas the calculations showed about 1.2 tons per sq. in.

It is well known that the problem of corrosion in the cargo tanks of oil-tankers is a serious one. Naval architects should, therefore, be interested in a unique approach to this problem in a paper entitled "A Structural Approach to the Problems of Corrosion in Tankers," read by Mr. D. B. Bannerman, before the American Society of Naval Architects and Marine Engineers, in November last. The paper describes a new type of structure which is intended to minimize corrosion. In essence, this consists of ensuring that all cargo tanks have smooth surfaces. To accomplish this, double plating is used for the bottom, side shell and all bulkheads, framing members being introduced between the double plates.

In this modified form of construction, a single longitudinal bulkhead on the center-line is employed. This double-plated center-line bulkhead is 4 ft. wide with the stiffeners on the inside. The double side shell is also 4 ft. wide, while the double bottom is 5 ft. deep. The stiffening is longitudinal throughout, and longitudinals are arranged on top of the deck. Also, the normal deck transverses, which are associated with the longitudinal system of framing, are on the outside.

A comparison between a tanker designed on this principle and a normal tanker, shows that cubic capacity is lost. Actually, the new arrangement gives a cubic capacity which is about 77 per cent of that of a normal vessel. While it would seem that a great deal of steel would be used in a structure designed in this way, calculations have shown that there is no significant difference between the structural weight in a vessel of this type, and one designed in the conventional way.

The great advantage of the proposed method of construction is that a large number of the structural features which tend to lead to corrosion troubles, are completely eliminated, and smooth surfaces are presented in the tanks which can, presumably, be very easily cleaned down. It might well be that this method of construction offers a new solution to the problem of tanker corrosion, and surely, if by the adoption of some such system, the life of the tanker is extended by only two or three years, it is worth while considering. It will be interesting to see if this proposed method of construction finds favor with owners, and its future development will be eagerly awaited.

New structural materials have an interest for the naval architect and the latest possibility along this line is reinforced plastics. An informative paper, entitled "Reinforced Plastics—a Structural Material for Marine Applications," was read by Mr. J. B. Alfers and Mr. W. R. Graner, before the American Society of Naval Architects and Marine Engineers, in May last.

Plastics, which were known before the Second World War, have become very well known in recent years. They are synthetic resinous products, and can be used for many purposes. The type which shows most promise as a structural material is that in which reinforcement is used, and in particular, that where fibrous glass is used as the reinforcing material. The strengths of these reinforced plastics depend very largely on the glass content, the strength increasing as glass content increases, and it is possible to produce materials which compare very favorably with other structural materials, including aluminum alloys and steel.

It would seem that, on the average,

the specific gravity of the materials is below 2, which certainly makes them worth considering from a shipbuilding point of view, since the weight which could be saved in their use to replace steel would be considerable.

So far as cost is concerned, reinforced plastics are more expensive, weight for weight, than steel or aluminum, and, at the present time, a figure of 45 cents per lb. might be expected.

In the United States of America, small plastic boats are quite common, and at the time this paper was written, a 57-ft. minesweeper was being designed and built of plastics. Other craft of a similar nature are contemplated.

To what extent these comparatively new materials will find their place in shipbuilding, only the future will show, but, clearly, the possibilities inherent in them should not be overlooked.

Turning to the subject of welding, there is comparatively little research work of a fundamental nature, in connection with the welding of steel in shipbuilding, to report as having taken place during the year. Readers will find much of interest, however, on the practical side of welding in a paper entitled "Trends in Modern Shipyard Welding," by Mr. R. J. W. Rudkin, B.Sc., published in the *British Welding Journal* for November, 1954. This paper discussed many important topics in connection with welding in the shipyards, and is worthy of study by all those who have to deal with the problem of the welded ship.

Another useful paper to be found in the *British Welding Journal* for October, 1954, entitled "A New Process in Stud Welding," by Mr. W. P. Van Den Blink, Mr. E. H. Ettema and Mr. P. C. Van Der Willigen, concerns a new process for carrying out stud welding. This process makes use of a cartridge for regulating the timing. The cartridge is a semi-conductor. It starts the arc between the stud and the plate, and also determines the distance between them. A simple stud welding gun, containing only a spring pressing the stud holder on to the workpiece, can be used.

In the field of welding of aluminum alloys, much of interest has been proceeding in the research work carried out by the British Welding Association. Several aspects of this work were dealt with in a paper entitled "Fusion Welding of Aluminum Alloys." Parts I., II., and III. of this paper were by Mr. H. E. Dixon, Mr. W. G. Hull, Mr. D. Adams and Mr. P. T. Houldcroft, and Part IV. by Mr. W. G. Hull and Mr. D. Adams, and were published in the October and November issues of the *British Welding Journal*.

A good deal of this work concerns aluminum alloys, which are outside the field of shipbuilding, as, for example, the welding of Duralumin-type alloys, and alloys of the Al-Zn-Mg-Cu type. Work is, however, being pursued on the metal arc-welding (flux shielded) of the alloy H.10, which is a heat-treatable alloy usable in shipbuilding. In addition, part of the program is concerned with the self-adjusting arc process, as applied to the Al-Mg alloys, especially of the N5/6 type. This, of course, has a direct interest for the shipbuilder.

In regard to the welding of aluminum alloys, some useful practical information was given in the issue of *The Shipbuilder and Marine Engine-builder* for August, 1954 (p. 474). This was an abstract of a paper entitled "Some Recent Developments in the Welding of Aluminum Alloys, and their future Applications in the Shipbuilding Industry," which was read by Mr. F. St. M. Brierly, B.Sc., and Mr. J. E. Tomlinson before the West of Scotland Branch of the Institute of Welding, in April last.

Of particular interest in this paper were the details given in connection with the construction of the all-welded aluminum-alloy yacht *Morag Mhor*.

This yacht is 72 ft. in length and has been entirely welded by the self-adjusting arc process. The plate material was of NP5/6 quality, while the extruded sections were NE6. Filler wire of NW6 quality was used. The successful welding of this yacht has shown that aluminum alloys can now be employed in large welded structures. A good deal of the work was carried out in the vertical, horizontal and overhead positions, and some difficult work in confined spaces has been favorably accomplished.

With the successful welding of structures, such as that of the *Morag Mhor,* a great deal of useful experience is being gained, and there is little doubt that, in the near future, the welding of aluminum alloys will be an accepted shipyard process.

RESISTANCE AND PROPULSION

A great deal of work has been carried out in this field during the year, and there has been another International Conference on matters related to these most important branches of the subject of naval architecture. This conference, which was the seventh to be held since 1933, commenced in Oslo in August, and was continued later at Gothenburg. As on previous occasions, the conference was preceded by a meeting, at which a number of papers were read on subjects related to the problems to be considered.

The official conference handbook has not, at the time of writing, been published, so that it is not possible to give any account of the decisions taken. We shall, therefore, have to content ourselves, in the meantime, by referring to the various subjects under discussion. These were the scale-effect on propellers and on self-propulsion factors, skin friction and turbulence stimulation, comparative propeller tests, sea-going qualities of ships, and presentation of resistance and propulsion data. It will be noted that the matters dealt with covered a very wide field, and are not only of interest to the research worker, but also to the practical designer, who has to design ships to fulfil certain conditions of speed and power. The full report will be awaited with interest. It is understood that there will be a conference in 1957, and that the invitation of Mr. M. L. Acevedo to hold it at El Prado, Madrid, was accepted by the committee.

The various papers which have been read during the past year on the research work carried out on resistance and propulsion indicate that the work may be divided into three categories, *viz.,* fundamental work on resistance, model experiments of the standard-series type, and ship-model correlation work and trial analysis. In the first of these categories, two papers may be mentioned, one by Mr. A. Emerson, M.Sc., entitled "The Application of Wave-resistance Calculations to Ship-hull Design," read before the Institution of Naval Architects in this country; the other was a joint paper by Mr. Ganett Birkhoff, Mr. B. V. Korvin-Kroukovsky, and Mr. Jack Kotik entitled "Theory of the Wave Resistance of Ships," read before the American Society of Naval Architects and Marine Engineers.

Mr. Emerson's paper deals with the application of wave-resistance calculations to hull design. The method generally adopted for determining the form of a ship to fulfil given design conditions is largely an empirical one, developed on the use and interpretation of accumulated ship-model data. Added to these data, the designer also has the mathematical solutions for

ideal forms; but, in spite of the progress made in the development of the theory, Mr. Emerson considers that a complete solution for the normal ship form is unlikely. He feels, however, that, if suitable empirical corrections can be made to the ideal solutions, not only will the study of wave-resistance be put on a firmer basis, but the nature of the corrections may lead to some simple alteration in the mathematical treatment to allow extension to more generalized hull forms.

Mr. Emerson discussed the correction factors which are necessary, and indicated the correction needed to bring calculated and measured resistance into reasonable agreement. He considers that similar corrections should be made to the mathematical treatment, and, if this is not possible, he states that correction factors can be obtained by studying the wave profiles of selected models.

In the second of these papers on ship wave resistance, the authors have examined critically the whole basis of the calculations. They begin by defining wave resistance and describe Michell's early work on this subject. This work was based on the assumption of zero viscosity in the fluid, and also upon the fact that the ship was slim. Approximations in calculations were also involved, in an attempt to produce numerical values for the resistance. In the present paper, the authors discuss the attempts to correct the errors involved due to approximations in calculation. Consideration is also given to the methods developed by Havelock, Guilloton and Weinblum to evaluate the Michell integral.

Two new methods are also presented, with the intention of simplifying and making more accurate the computation and, at the same time, giving an idea of the relation between the ship form and the wave resistance.

While it would be true to say that an accurate computation of ship resistance must still be far away, if methods such as those suggested in this paper can indicate how the resistance will vary with the various features of the ship form, a great step forward will have been made. In addition, if calculations can assess the relative importance of the different features of the ship form, then much will have been gained.

The study of the frictional resistance of surfaces has been greatly assisted by the study of the problem of the boundary layer, and a further paper on this subject has been presented to the American Society of Naval Architects and Marine Engineers by Mr. Francis R. Hama, entitled "Boundary-layer Characteristics for Smooth and Rough Surfaces." The work described in this paper is, first of all, a review of boundary-layer developments on smooth plates with zero pressure gradient. An account is given of measurements made in a wind tunnel of the velocity distribution along a plate covered by a series of artificial roughnesses of controlled characteristics. The analysis of these experiments has revealed that two universal laws—the wall law and the velocity defect law—and an intermediate region common to the two, in which the velocity profile is logarithmic, are essential features of the turbulent boundary layer. The universality of the roughness effect, regardless of outside flow conditions, which has been conjectured for a long time, has now been substantiated by this investigation, at least for the case of fully developed roughness action at zero pressure gradient. The nature of the effect of the roughness pattern still requires systematic investigation. A further conclusion is that the pronounced increase in surface resistance with certain paints, as reported by Dr. Todd, is within reason.

Research work in this country on the subject of frictional resistance has not lagged, and further work by Mr. G.

Hughes, D.Sc., Ph.D., has become available during the past year in a paper, which he read before the Institution of Naval Architects, entitled "Friction and Form Resistance in Turbulent Flow, and a Proposed Formulation for use in Model and Ship Correlation." Dr. Hughes' latest work gives the result of further experiments on the turbulent resistance of smooth surfaces in the range of Reynolds numbers from 0.03×10^6 to 250×10^6, and covers a range of length-breadth ratio from nearly zero to 64. For this purpose, submerged plates and sheets were used, and also shallow-draft pontoons. From the result of these experiments, a new formula for frictional resistance has been produced, which is as follows:—The specific resistance coefficient:—

$$C_f = R_f / \tfrac{1}{2}\rho S v^2 = 0.066(-2.03 + \log_{10} R_n)^{-2},$$

where R_n is the Reynolds number. This formula is for the basic line obtained from the experiment in which the aspect ratio is infinite. The lines for other values of the length-breadth ratio are at constant percentages above the basic line. Experiments carried out with submerged cylinders showed that the effect of transverse curvature was to steepen the slopes of the friction lines for the plane surfaces. It is considered that accurate extrapolation to cover all sizes of ships is possible in the light of this experimental work, and the formula which it has produced. A conclusion which has been arrived at from this work is that the hull resistance of the ideally smooth ship is much less than has been supposed hitherto, and it is doubtful whether the best result which has been achieved on trial is within reach of the best possible results. It is suggested that it is probably impossible to achieve these results without a special covering to the hull surface similar to the plastic which was used in the model tests.

While on the subject of minimum resistance of surfaces, mention should be made of an interesting piece of research work carried out by Mr. R. L. Townsin, B.Sc., and recorded by him in a paper on "Experiments with a Low-drag Hydrofoil," read before the Institution of Naval Architects in September. The hydrofoil was of symmetrical shape, and was designed and manufactured so as to produce a laminar-boundary layer over 60 per cent of its surface. The drag of the hydrofoil was measured in a towing tank with various boundary conditions and at zero angle of incidence; stream-line observations were made for all conditions of the hydrofoil. Calculations of the drag of the hydrofoil were made at zero incidence for various Reynolds numbers and transition points. It was found in the experiments that it is possible to produce an extensive and controlled laminar boundary layer in the towing tank, and to obtain, also, the low drag associated with laminar flow. Small excrescences from the surface of the hydrofoil are, however, likely to disturb the laminar boundary layer, and once the flow was tripped, it was observed that it could not be made to join the hydrofoil as a laminar layer again.

In view of these experiments, one is likely to inquire to what extent is it possible to achieve laminar flow on a ship hull. Mr. Townsin's remarks on this subject seem to indicate that a very high degree of smoothness would have to be achieved on a ship hull, in order to approach the ideal of laminar flow. Moreover, even if it were possible to obtain this degree of smoothness, the action of fouling would, even in the early stages, produce sufficient roughening to disturb the laminar flow. It would appear, however, that the situa-

tion is rather brighter in connection with the marine propeller, where a high degree of finish is already achieved. There may then be some future for propeller-blade sections which will stimulate laminar flow, and thus reduce blade resistance and hence, improve propeller efficiency.

Turning now to the field of model experiments on resistance of ship forms, a considerable amount of work carried out for the British Shipbuilding Research Association has been published during the past year. This has appeared in two papers before the Institution of Naval Architects. The type of form chosen for these experiments was one having a block coefficient of 0.65 and the features of the form which were investigated were, first of all, the longitudinal position of the center of buoyancy and the bilge radius. Other features, whose influence on resistance was examined, were the breadth-draft ratio, and the length-displacement ratio. Three positions of the center of buoyancy were investigated, *viz.*, ½ per cent L forward of amidships and ½ and 1½ per cent aft. Bilge radii of 4, 6, 8 and 10 ft. for a 400-ft. ship were tested. The other features of the form were a breadth of 55 ft. and a draft of 26 ft. The resistance experiments showed that the influence of bilge radius was small, while the speeds at which the value of © rose steeply for the three positions of center of buoyancy tested were given by $V/\sqrt{L}=0.75$, 0.80 and 0.85, respectively. It could be concluded, therefore, that these speeds represented the economical speeds for those positions of the center of buoyancy. Propulsion tests were also carried out on the models, but with one bilge radius only. It was found that the influence of position of center of buoyancy on the quasi-propulsive coefficient was small.

The second part of this work on the 0.65 block-coefficient forms was in two parts. In the first, a range of breadth-draft ratio was tested with constant displacement and length, and, in the second, a range of length-displacement ratio was tested for a constant breadth-draft ratio. The breadth-draft ratios tested were 2.12, 2.52, 2.96 and 3.43, while the values of length-displacement ratio, as measured by $L/\nabla^{1/3}$, were 6.34, 5.56, 4.98, 4.74, 4.525 and 4.33. The experiments on variation of breadth-draft ratio showed that resistance increased with increase in this ratio, but at about $B/d=3.0$, there was a tendency for the resistance to decrease again. The experiments with varying length-displacement ratios showed that, with decrease of $L/\nabla^{1/3}$, resistance generally increased, and this was more marked at the higher speed-length ratios. The increase in resistance was not, however, found to be of a regular character over the range of length-displacement ratio tested.

As a continuation of the work on coasters, a series of propulsion tests carried out at the National Physical Laboratory have been published by Mr. J. Dawson, B.Sc., in a paper entitled "Resistance and Propulsion of Single-screw Coasters, Part II.—L/B=6," read before the Institution of Engineers and Shipbuilders in Scotland. In that series of experiments, the effects on propulsion of block coefficient, position of center of buoyancy, and speed were examined. Only one propeller was used, and all the experiments were carried out at constant draft. Curves are given in this paper, showing the variation of QPC with revolutions per minute for three different stern arrangements, *viz.*, unbalanced stream-lined rudder with vertical or shaped fin, unbalanced single-plate rudder with shaped fin and unbalanced stream-lined rudder with square stern-post. The former of these appears to give the best propulsive coefficient, being some 7 or 8 per cent better than the others. Wake

and thrust deduction factors are also given, plotted to a base of block coefficient for the three types of stern arrangement mentioned.

In the realm of full-scale ship-resistance experiments, it was of great interest to have made available during the past year additional information on the *Lucy Ashton*, in a paper by Mr. H. Lackenby, M.Sc., entitled "B.S.R.A. Resistance Experiments on the *Lucy Ashton;* Part III.—The Ship Model Correlation for the Shaft-Appendage Conditions," read before the Institution of Naval Architects, in September last.

This paper dealt with experiments which were made with the shaft appendage in position. Two types of appendages were used, *viz.*, the normal twin-screw bossings generally found in merchant ships, and the A-bracket-type of shaft appendage. In the case of the full-size ship, the appendages were, of course, dummy, since the ship was originally a paddle vessel, but representative types of these were constructed from welded sheet metal. The resistance increments due to fitting these types of appendages were measured, and a similar procedure was adopted for the models, which varied in length from 9 ft. to 30 ft.

It was found that, with bossings, the resistance increment was subject to scale-effect. This effect varied from zero at the lower speeds, and became quite appreciable at the higher speeds. It is of interest to note that over the ship speeds from 8 to 12 knots, and for the usual model length from 12 ft. to 20 ft., the ship increments were roughly one-half of those measured on the model.

With regard to shaft brackets, the scale effect did not show any tendency to vary with speed, and, once again, the percentage increase obtained on the ship was about one-half that determined from the model experiments.

Concerning the relative merits of bossings and shaft brackets, it was found that the latter offered rather less resistance at speeds between 8 and 13 knots. At other speeds, the position was reversed. Ship-model correlation throughout these experiments was carried out on the basis of the Schoenherr line. The derived roughness allowances in both the bossing and shaft-bracket conditions, showed a good measure of agreement, not only among themselves but also with those for the original naked hull condition.

A very valuable series of experiments have been carried out in Holland, the details of which are to be found in a paper entitled "Scale-effect Experiments on 'Victory' Ships and Models." The scales of six of the seven models tested were one-sixtieth, one fiftieth, one-fortieth, one-thirtieth, one-twenty third, and one-eighteenth full-size, while the seventh model was a large-scale one, one-sixth full-size. This last mentioned model, which was over 22 metres (72 ft.) long, was called *D. C. Endert, Jr.* The model was built of steel, and weighed, when empty, about 30 tons. The propulsion equipment in this model boat, consisted of a Ford 43-H.P., four-cylinder, gasoline engine, which drove a D.C. generator, and this, in turn, drove a D.C. motor coupled to the propeller shaft by a 4:1 reduction gearing. The boat was fitted with thrust and torque measuring equipment, and also had a resistance dynamometer for measuring its resistance when being towed. This model boat was tested in the waters off the Croatian coast, under ideal weather conditions.

The resistance and propulsion results for this rather unique large-scale model were correlated to the actual ship, and the family of smaller models which were tested in the towing tank. Three different methods were used for the purpose of correlating the results, these being based on the Schoenherr, the Hughes, and the Lap-Troost extra-

polators. Readers are referred to the original paper for details of this most interesting series of tests, but we shall refer here briefly to the results and conclusions so far achieved.

The experiments carried out to date would not apparently suggest the correctness of any one of the means of extrapolation used, and it was discovered that the Schoenherr line is too flat. An attempt to determine a thrust extrapolator instead of a resistance extrapolator did not prove successful, the trouble being attributed to the inaccuracy of the thrust measurements. It would appear from the experiments that the surface of the model boat could not be regarded as hydro-dynamically smooth, and it is suggested that, in order to obtain a correct assessment of the roughness, tests should be carried out with flat plates which have the same type of surface finish as the model boat.

Another interesting piece of analysis in the ship-model correlation field has been carried out by Professor J. W. Bonebakker, the results of which are recorded in a paper entitled "On Collecting Ship Service-performance Data and Their Analysis," read before the North-East Coast Institution of Engineers and Shipbuilders. We had occasion to refer to Professor Bonebakker's work in this field on a previous occasion, and in the present instance he has dealt with model experiments, trial results and service performance data for a single-screw tanker. Based on an analysis of actual performance of the ship, the author shows how surface allowances can be made on power for any given weather conditions.

Further interesting research work, carried out by the Netherlands Ship Model Basin, Wageningen, has been communicated in the form of a paper entitled "Correlation of Model and Ship Trials of a Shallow-draft Rhine vessel," by Professor Dr. Ir. W. P. van Lammeren and Dr. Ir. J. D. van Manen, read before the North-East Coast Institution of Engineers and Shipbuilders. The vessel in question was the tanker *Arabia,* and was 76.5 meters by 10.0 meters and had an even-keel draft of 1.85 meters. Tests with this ship showed that it is not possible to obtain the speed of a ship by direct measurement in restricted waters, largely because it is constantly being accelerated and decelerated by the irregular flow phenomena in the river.

With regard to model experiments in shallow water, with which the ship tests were related, the influence of the tank width increases as the depth of water decreases. The fact that the resistance curves for one depth of water and varying tank widths coincide, when plotted against the number $\frac{V}{\sqrt{g R_h}}$, where R_h is the hydraulic radius, provides a simple and valuable basis for the elimination of the wall effect in model tests. The experiments described in the paper lead to some interesting conclusions for future work. For example, it would seem desirable that tests on the *Arabia* should be carried out in still water of varying depths. On the model side, it would be useful to have comparative tests with a model in the tank and in a flow-tank. Also experiments in the tank with water of limited depth and with different widths of tank should yield important information as to the influence of width on model results from vessels in restricted water. It is further suggested that experiments on a family of models would assist in the extrapolation problem for vessels of this type.

The ship model correlation problem is very much in the minds of experimenters in all countries and some aspects of this matter were dealt with by Mr. J. F. Allan, D.Sc., and Mr. H. J. S. Canham in a paper entitled "Ship Trial Performance and the Model Prediction," read before the Institution of

Naval Architects during the past year. The work covers a very large amount of analysis of data collected and studied by the National Physical Laboratory and the British Shipbuilding Research Association. Some points which emerge from this work are, first, that the greatest care must be used in connection with data collected from trial trips, if these are to be of any value. Another point is that there is a significant difference between the results for a riveted and a welded ship, a completely welded ship showing some 15 per cent reduction in power for the same speed, as compared with the riveted ship. The trial results for the class of ship dealt with in the paper are some 15 per cent below the E.H.P. which would be predicted by Froude, and about 5 per cent below the Schoenherr prediction, but considerably above the E.H.P. by the new method proposed by Hughes. There are also indications that the actual roughness of a clean painted surface varies very greatly, and this matter deserves special study. It is quite clear from researches such as this, that the prediction of ship powers from model experiments is a complex problem, and that much study is still required in order to put this matter on a sound basis.

Naval Architects will remember the resistance experiments carried out in America on single-screw ship forms, referred to as Series 60. Reference to this work was made in our equivalent article for 1953. Readers will be interested to know that the results of propulsion experiments are now available for this series of models, and that these data are to be found in a paper read before the American Society of Naval Architects and Marine Engineers. In this series of propulsion tests, the factors investigated included propeller diameter, block coefficient, displacement and trim. A set of parent propellers developed from the Troost B-4 series, with diameters equal to 70 per cent of the draft, were constructed. It is quite impossible to discuss the results of these tests here, it is sufficient to say that a great deal of valuable information is to be found in the paper which will well repay careful study by ship designers.

DESIGN OF SHIPS AND GENERAL TOPICS

In the field of design of ships, a vessel of note which was completed late in 1953 was Her Majesty's Yacht *Britannia,* and some comments on this ship seem appropriate. The matter was dealt with fully by the Director of Naval Construction, Sir Victor G. Shepheard, K.C.B., R.C.N.C., in his paper entitled "Her Majesty's Yacht *Britannia,*" read before the Institution of Naval Architects in the spring of 1954. One of the bases of design of this vessel was that of making it a dual-purpose ship, *viz.*, that she should function as a Royal Yacht in peacetime, but be capable of being converted into a hospital ship in time of war.

It is of interest to mention that the design of this ship was considered before the last war, outline drawings having been submitted to various shipbuilders during 1939. The advent of hostilities, however, held up further consideration of the matter, and it was not until 1951 that the project was reconsidered.

The dimensions of the ship are 360 ft. B.P., by 55 ft. breadth molded, by 32 ft. 6 in. depth molded to upper deck. The ship has a load displacement of 4,715 tons and a speed of 22.75 knots on trial, with the propelling machinery developing 12,000 S.H.P. The design of the hull form was the subject of model experiments both by Messrs. John Brown & Co. (Clydebank), Ltd.

(the builders of the ship), and at the Admiralty Experiment Works, Haslar. Both smooth-water conditions and rough seas were considered in attempting to arrive at the most suitable form. It is of interest to note that the form finally decided upon had a slightly bulbous bow—a feature not generally found in British ships. The superstructure and funnel of the ship were the subject of wind-tunnel experiments, in order to ensure a draft-free bridge and to avoid smoke coming down on deck. Among the many items of modern equipment in this ship are the stabilizer fins, designed to be capable of reducing a roll of 20 deg. out-to-out to 6 deg. out-to-out, at a speed of 17 knots.

In the merchant-ship field, some very useful information has been collected, and made known, on the operation of the "Mariner"-type ships. This is to be found in a paper entitled "Operation in Service of the 'Mariner'-type ship," written by Mr. William G. Allen and Mr. E. Kemper Sullivan, and read before the American Society of Naval Architects and Marine Engineers. We have referred to the design of these very interesting vessels in previous years, and it is extremely useful to have made available now, data on their operation. The service performance discussed in this paper concerns the transportation of military cargoes. As a result of this experience, several design changes were found to be desirable and have been effected. It is revealed that, in a period of 17 months, 20 ships covered over 1,300,000 miles at an average speed of 20.17 knots. It will be found that all aspects of the operation of the ships, including running costs, are dealt with in this very informative paper.

The design of another type of ship has been dealt with by Mr. W. A. Stewart, in a contribution to the transactions of the North-East Coast Institution of Engineers and Shipbuilders. In this paper, entitled "The Development of Ore-carriers in Recent Years," the author has given details of some of the most modern vessels of this type. The increase in size of these ships over the years is interesting. Some 35 years ago, the average ore-carrier had a deadweight of between 9,000 and 12,000 tons. The modern vessels of the *Bomi Hills* and *Hanna* types vary in length from 600 ft. to 756 ft., and have deadweights from about 24,000 tons to 59,000 tons. The ships of the type discussed in this paper have wing tanks, with ore holds on the center-line. The tanks can be used for water ballast, or, alternatively, they can be employed for the carriage of cargo oil, so that these ships can be regarded as having a dual purpose.

It is of interest to note the high stresses which can exist in the structures of ore-carriers. In the *Bomi Hills* type, for instance, with a displacement of over 30,000 tons in the oil-condition, the compressive stress in the deck, calculated for the standard sagging condition, amounts to no less than 9.4 tons per sq. in. The type of structure employed in the modern ore-carrier consists of longitudinal framing with deep transverse, not unlike that employed in oil-tanker construction; but, in the case of the ore vessel, a deep double bottom is incorporated, this space being used for the purpose of water-ballast tanks.

Yet another type of vessel dealt with during the year is the trawler, and we would refer readers to a very interesting paper, entitled "A Note on Large Trawlers," by Professor Ir. W. H. E. Jaeger, read before the Institution of Naval Architects. This paper considers the design of trawlers of 165 ft. to 185 ft. in length, with displacements of from 1,000 to 1,300 tons. The design of ships of this type is considered in some detail in the paper, and a general-arrangement plan is given for a deep-freeze trawler. The vast difference in

appearance of these vessels which has taken place should be noted, the modern trawler looking very unlike the traditional type. Because of the varying conditions of loading when the trawler is fishing and when running at full speed, two different approaches have been adopted to make the best use of the power available. These are to employ a part of the propulsion power for driving the trawl winch, and, alternatively, to use a controllable-pitch propeller. It would appear that the controllable-pitch propeller offers the best solution to the problem.

Turning to some miscellaneous topics in connection with shipbuilding, the problem of cathodic protection is still to the fore. During the past year, Mr. J. Lamb, O.B.E., with Mr. E. B. Mathias, B.Sc., and Mr. W. Godfrey Waite, read another paper on the subject of using this method in connection with the protection of the interior of cargo compartments in oil-tankers. This paper which was entitled "Cathodic Protection of the Interior of Cargo Compartments in Oil-tankers," and was read a few months ago before the North-East Coast Institution of Engineers and Shipbuilders, reviews further tests which have been made with this method of protection. In the tests described in this paper, a five-year-old tanker, the *Auris*, was chosen. This was a 12,250-ton ship which had spent five years in carrying "white" cargoes and was due to change over to "black" oil. The vessel was fitted-out with the cathodic protection system in October, 1952, and was finally examined and the results assessed about a year later. It appears that the cathodic system can be 100 per cent successful, if it is applied continuously. The suggestion has been made that, in crude-oil ships, where comparatively little corrosion occurs in non-ballast compartments, corrosion might be rendered non-existent by designing a number of small compartments to carry ballast only, while the majority of the other compartments carry cargo oil only. It is interesting to note, finally, that the cathodic-protection system produces no effect on the cargo itself, analysis of cargo showing a magnesium pick-up of only two or three parts in a million.

The stern-first method of launching ships is so common in this country that one rarely thinks of other means of setting ships afloat. The sideways launching method is, however, sometimes employed, and it will be of interest to readers that we refer to a paper recently read by Mr. D. J. Doust, B.Sc., entitled "The Side Launching of Ships — with special reference to Trawlers," before the Institution of Naval Architects. This deals with an investigation on the subject, carried out on the model scale at the National Physical Laboratory. These results have been linked with observations on full-scale launches, by fitting gyroscopic equipment on ships just prior to the launch, the equipment being wired to recording units on shore. The investigation, which was particularly aimed at the trawler type of ship, has yielded some practical results, in that a method has been developed enabling the launching characteristics for side launches to be predicted with practical accuracy, using design diagrams. Further experiments are envisaged for other ship types, with a view to extending the range of the diagrams.

Finally, we would refer to an investigation carried out in the United States of America on the propeller forces exciting hull vibration. The results of this work have been made known in a paper read before the American Society of Naval Architects and Marine Engineers, by Professor Frank M. Lewis and Mr. A. J. Tachmindji entitled "Propeller Forces Exciting Hull Vibrations." This work was carried out under the sponsorship of Panel H-8 of the American Society. It

was principally concerned with the propeller forces arising in single-screw ships. Experiments were carried out on models, the factors investigated including revolutions per minute, axial and propeller tip clearances, presence of rudder, and rudder-propeller clearances. Comparison was also made with observations taken on full-size ships. In this comparison, it emerged that if it were not for the different characteristics of the wake belt for model and ship, it would be possible to compare the forces on the basis of Froude's law. Further model experiment work is envisaged, in which the influence of various parameters will be examined, such as hull, propeller and propeller-rudder clearances, number of blades, skew and various hull characteristics, such as the influence of the immersed area over the propeller, length of overhang of stern, and "U" *versus* "V" sections.

A BENCH TESTER TO INVESTIGATE BASIC SOURCES OF GEAR NOISE

R. C. JOHNSON

(Author's biography on page 565)

INTRODUCTION

Machinery noise, which can vary from a minor irritant to a major factor in the health and efficiency of most machine operators is literally a matter of life or death to Naval submarine personnel in wartime.

Prior to World War II, the U. S. Navy's concern with machinery generated noise was concentrated on airborne noise. The increasing use of submarines and underwater listening devices has focused attention on waterborne noises, radiated by surface ships as well as submarines. Rotating machinery in general and main reduction gears in particular are major sources of this radiated noise. During World War II, various treatments were tried, without much success to quiet gears, until finally propulsion geared drives in most submarines were replaced by large, heavy, slow speed, direct drive motors. However, the increasing use of high speed prime movers, particularly since the advent of gas turbines and nuclear powered steam turbines can only mean the return of reduction gearing. To couple high speed machinery operating as high as 20,000 rpm with propellers operating from 250 to 400 rpm, it is evident that reduction gears are necessary.

The Navy has various gear noise investigations now underway as to the influence of materials, tooth geometry, tooth finish, lubrication and other factors, on noise generation. Much progress has been made. However due to the complexities of the problem and the size and cost of the components involved, test work is necessarily slow and expensive.

It is generally agreed that increased accuracy, and better tooth surface finishes can help alleviate gear noise. Many theories and conclusions on the subject are directly contradictory and

oppose findings from controlled laboratory tests. It is apparent that a basic knowledge and determination of the fundamental causes of gear noise is extremely necessary, to speed the solution of this problem.

As a part of the determination of basic causes of gear noise, the Bureau of Ships became interested some time ago in the contribution to noise of the air and oil in the tooth mesh, as well as the action of the individual teeth coming into and leaving the mesh. To study this problem in detail, a medium sized bench tester was conceived. This apparatus may seem large in comparison with most bench testers, yet compared to full size marine gears which may be as large as 100 inches in diameter, the apparatus is small, yet not so small that the basic characteristiccs of full size gears are lost. In addition, the test gears are of the same center distance as those in a 300 KW test stand at the U. S. Naval Engineering Experiment Station at Annapolis, Maryland where most of the gear noise test work has been done. It is hoped to obtain some correlation between these two tests.

OBJECTIVES

Some of the objectives of this investigation are listed below, others are expected to be determined as the test program progresses.

a. Determine the air and structure borne noise generated by the displacement of air, resulting from a gear tooth meshing with another tooth.

b. Determine the air and structure borne noise generated by the displacement, pressurizing, cavitation, turbulent flow and rapid movement of lubricant along the teeth and in the space surrounding the gears.

c. Study the individual tooth in the gear mesh and its deflection and possible vibrations with or against other teeth and the subsequent generation of air and structure borne noise.

d. Devise means for eliminating or isolating noise resulting from the above.

BACKGROUND

To substantiate the theory that the displacement of air, caused by the meshing of one tooth between two other teeth, may cause air-borne or structure borne noise, the following example is given. Data used are from the 300 KW sets on test at the Engineering Experiment Station.

Gear Data

	Gear	Pinion
R.P.M.	1200	6000
Dia. Pitch.	10	10
Nr. Teeth	247	50
Helix Angle	9° 2′ 15″	9° 2′ 15″
Face Width	6 inches	6 inches
Pitch Dia.	25.009 inches	5.063 inches

When operating at full speed of 6,000 pinion r.p.m., the following conditions will be found.

a. *Frequency of tooth contact mesh*

Nr. Teeth × R.P.M. = 50 × 6000 = 300,000 per min. or 5000 cycles per second.

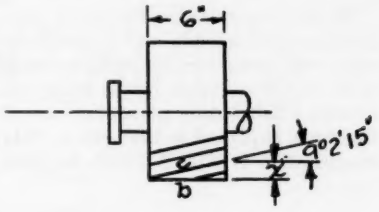
b. *Velocity of travel across face.*

Pitch Line Velocity = π × Pitch Dia. × R.P.M.

$PL_V = \pi \times 5.063'' \times 6000$

= 95,460 inches/min.

= 1591 inches/sec.



$$X = b \times \text{Tan } A = 6 \times .15905 = 0.9543 \text{ inches}$$

$$C = \frac{b}{\text{Cos } A} = 6 \div 0.98759 = 6.075 \text{ inches}$$

At PL_V of 1591 in./sec., pinion travels "X" distance in .0006 seconds, and at the same time, air or oil bubble travels across face of helix, or "c" distance of 6.075" in same time, or at a rate of 10,125 inches per second or 843 ft. per second.

c. *Volume of air or oil displaced by meshing teeth at 6000 rpm, in one complete revolution.*

Volume ($ft.^3$ per min.) = Area of tooth × Face Width × No. of Teeth contacts per min. ÷ 1728

$$\text{Vol.} = \frac{0.0293 \text{ in.}^2 \times 6.075'' \times [50 \times 6000]}{1728} = 30.9 \text{ cu. ft./min.}$$

From the above calculations, it can be seen that the tooth mesh frequency is well within the audible range of 30 to 20,000 cps and right in the most sensitive range of the ear, which is 500 to 6000 cps. The volume of air displaced, 30.9 cu. ft. per min. or 892 cu. inches in a second at 5000 cps may result in appreciable noise. The change in air pressure resulting from the movement of air from between two teeth will depend on a number of factors including clearance between teeth, velocity of meshing elements and change in openings at either end of the three teeth that tend to form a trap for the air. A series of curves showing change in velocity of travel across the face of the teeth, against helix angle and volume of air displaced per unit of time, against helix angle are shown in Figure 1.

d. It is of interest to note, that the smaller the helix angle, the greater the velocity of travel across the face. With a helix angle of 0° 15' 0", the velocity of travel across the face is approximately 30,000 ft. per second which is almost thirty times the speed of sound in air at sea level, at approximately 1,000 ft. per second.

The rapid movement and build up and release of pressure in the lubricant between the meshing teeth is a second possible source of noise. There are several conditions existing in the oil between meshing teeth that may contribute to noise. With a rapid release in pressure that is built up in the film, small particles of gas may unite around a nucleus and expand to form a bubble which generates sound. The collapsing of such bubbles will also probably result in noise. A second source of hydrodynamic noise is the possibility of actual cavitation in the oil in the meshing teeth. A third possible source is the impact of

oil on the gear teeth case after being discharged from the trap formed by a tooth moving into mesh between two other teeth.

To investigate these factors, a preliminary test agenda and a specification was prepared in collaboration with the New York Naval Shipyard Material Laboratory personnel, where testing is expected to take place. This specification is MIL-A-17093. In June of 1952, a contract was negotiated with the Falk Corporation of Milwaukee, Wisconsin, for the first phase of this investigation, a design study of the apparatus. This phase is now being completed and construction is due to start soon.

GENERAL DESCRIPTION

The layout of the test rig as designed is shown in Fig. 2. The machine basically consists of a variable speed D.C., 75 HP electric motor drive, a brake, clutch, flywheel, drive gear unit with torque applier and test gear unit. The test gears are of equal diameter and of equal number of teeth. These gears are driven by the electric motor through a set of drive gears of the same diameter. The "locked-train; back to back" method will be used for loading the test gears, with the electric motor used for overcoming starting torque, and making up frictional losses. A hydraulic torque applier located on the end of one of the drive gears will apply torque by displacing one of the helical drive gears axially, relative to the other, thus winding up or torqueing the shaft to a pre-determined load. Torque can be applied and varied while the machine is in operation. Specifications call for the torque applier to be capable of applying torque from zero pound inches to a maximum of 10,000 pound inches. The motor is to be capable of operating at variable speeds from 1000 rpm to 3000 rpm. A recording torque meter will measure torque applied to the test gears. Test gears are to be completely isolated to avoid transmission of any noises from drive gears and other parts of the machine.

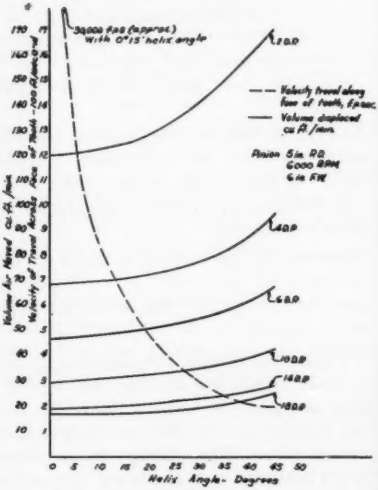


Figure 1. Volume of Air and Velocity of Travel vs. Helix Angle.

SPECIFICATIONS— OR DETAILED DESCRIPTION

Details of various components as specified and designed are related in the following paragraphs.

a. Driving Motor

The motor is to be of sufficient size to operate at variable speeds from 1000 rpm to 3000 rpm and have sufficient power to start the test gear unit with 10,000 pound inches of torque applied. A 75 HP, 1000 to 3000 RPM, 230 volts D.C., open self ventilated frame motor, built to commercial specifications

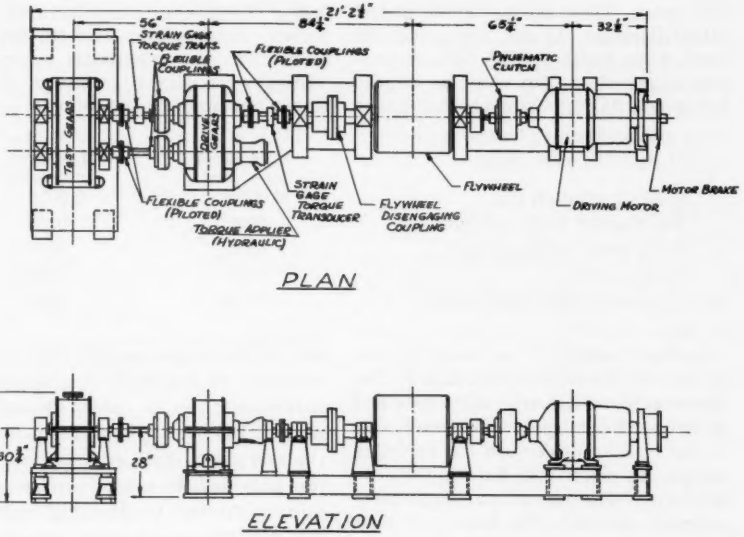


FIG 2 - LAYOUT OF TEST RIG

by Allis-Chalmers Manufacturing Company was chosen.

b. Brake

The drive motor to be provided with a brake to bring the motor to rest within 10 seconds after the clutch is disengaged. This is accomplished by a pneumatic brake, Falk Corporation, Airflex 10-4B.

c. Clutch

The clutch is to be a quiet operating unit which can quickly be disengaged. It will operate at speeds up to 3000 rpm and transmit the maximum power required by the gear test machine. A pneumatic clutch, Falk Corporation "Airflex 10-4B" is used. This will allow for quick disconnection of the motor. A lockout device is provided to permit operation of the motor, without engaging the clutch.

d. Flywheel

The flywheel has sufficient mass so that when the machine is operated at 3000 rpm with zero torque on the gear shafts, and the clutch is suddenly disengaged, the energy stored up in the flywheel is sufficient to drive the gears at a uniform deceleration no greater than 50 rpm with the gears submerged in oil. A flywheel, approximately 30" dia., by 30" long is to be used. These dimensions are tentative, due to the fact that actual HP required has not been determined.

e. Flywheel Disengaging Coupling

A "Fast" Nr. "5 cut out coupling" is provided forward of the flywheel to permit its disengagement from the system if desired.

f. Drive Gears

The drive gears are necessarily of the same center distance, 15 inches, and the same ratio, 1 to 1, as are the

test gears. They are to be 15 inches pitch diameter, 12 dia. pitch and 180 teeth. Face width is 6⅞". The accuracies required are the same as for the test gears, AGMA Class 4 tolerances or better. To give the manufacturer some leeway close tolerances were set as a goal, then the maximum acceptable errors were slightly relaxed. Tolerances required are as follows:

	Max. Error as a goal	Max. Error acceptable
Runout of pitch dia.	0.0005"	0.001"
Pitch error tooth-to-tooth	.00015"	.0002"
Pitch error accumulated	.0005"	.0008"
Profile error	.0001"	.0003"
Circumferential lead error	.00015"	.0002"

Surface roughness of teeth is not to exceed 15 micro-inches R.M.S. The drive gears were originally specified as double helical gears. However, due to the method developed for applying torque, as described below, this requirement was waived and single helical gears allowed. The drive gear case is to be of a sandwich type construction, with damping material attached to it, to prevent resonance, and amplification of noise.

g. Torque Applier

The torque applier is required to be quiet operating, and capable of applying torques from 0 pound inch to 10,000 pound inches. A hydraulic torque applier was developed. This is mounted on the stub end of one of the drive gears. The stub shaft has a thrust collar mounted on it with an actuating bellows on either side. By filling either or both sides of the bellows with oil, an axial movement of the one gear relative to the other up to .440 inches in either direction, can be achieved. This axial movement, relative to the stationary gear and shaft, causes a windup between the shafts, and imposes a precalculated torque on the drive gears, regulated by the amount of axial movement. Thus, torque in varying amounts can be imposed during actual operation of the gears. Reference points are provided on the outside of the torque applier, with a small indicator on the shaft, so precise axial movements can be achieved and regulated. Between the drive gears and the test gears, Falk Airflex Couplings are provided, to prevent noise transmission through the shafting, and flexible piloted couplings are provided to allow for axial movement. On the shaft between the flywheel and drive gears, a Baldwin-Lima-Hamilton Corporation Model B-5 Torque Meter of 5000 inch lbs. capacity is provided to measure torque required to rotate the system. Between the drive gears, and test gears, on the same shaft and opposite to the shaft with the torque applier, a Baldwin Model B-12 torque meter is provided to measure the torque applied to the test gears, by the torque applier. Both of these torque meters are equipped with recording instruments.

h. Test Gears

Test gears are of the same size and accuracies as are the drive gears. However, the test gears are 10 diam pitch, with 150 teeth and a 4 inch face width. These gears are double helical. Four sets of test gears are to be provided. Each set will consist of three gears; two of steel and one of a nonmetallic material having high damping properties, such as Micarta. The four

sets vary slightly as shown below:

Set Nr. 1

P. Dia. 15 inches
Helix Angle 0 degrees (spur)
Pressure angle—14½°
Backlash 0.010″—0.012″

Set Nr. 2

P. Dia. 15 inches
Helix Angle 9°8′15″
Pressure angle—14½°
Backlash 0.010″—0.012″

Set Nr. 3

P. Dia. 15 inches
Helix angle—to be selected later
Pressure angle 14½°
Backlash 0.010″—0.012″

Set Nr. 4

P. Dia. 15 inches
Helix angle 9°8′15″
Pressure angle—14½°
Backlash 0.030″

The affect of backlash, and helix angle on noise output will be studied with these gear sets.

i. Gear Case for Test Unit

The gear case for the test gear unit is required to withstand a maximum internal fluid and air pressure of 25 p.s.i., absolute and a vacuum of 25 inches of mercury. This is to enable tests to be run under conditions varying from the case nearly full of oil, to the case being under a vacuum. Thus it is hoped to determine the affects of too much and too little oil or air on noise generated. The case is to be basically of steel construction with patch sandwich plates of asbestos and steel, for damping purposes, held in place by studs. Large openings covered by "Herculite" tempered plate glass, are provided on the top, bottom, sides and ends of the case for visual observation and high speed photography of the gears under test. The test gear case is supported separately from the rest of the test apparatus, and to be mounted on "Lord" isolation mounts.

j. Lubrication

The gear test apparatus is to be lubricated by two different oil systems. The purpose of this is to prevent any noise or vibration from being transmitted to the test gears from any other part of the apparatus through the oil lines.

One system will supply filtered oil to all bearings and the drive gears. This system consists of a DeLaval Nr. A322A-187 oil pump, driven by a 5 H.P., 1750 R.P.M. motor, a 4 pass Kewanee-Ross Nr. BCF803 oil cooler, an oil heater with 3 Calrod 5 K.W. elements and a Purolator Nr. G-141J-13 oil filter, all mounted on a separate stand and isolated from the gear housing.

The second system, for the test gears only consists of a DeLaval A313A-143 oil pump, driven by a 2 H.P.-1150 rpm motor; a Kewanee-Ross Corporation oil cooler and an oil heater with six 7½ KW Calrod elements and a Purolator "Michonic" P92-02 oil filter and a Fischer and Porter non-recording flow-meter. This system is separately mounted and isolated from the test gears.

For the first system, oil is to be provided at operating temperature within 30 minutes after starting and maintained at this temperature during operation. Remote gages and thermometers are provided to measure the following:

(1) Temperature of oil into heat exchanger.

(2) Temperature of oil from heat exchanger.

(3) Temperature of oil at manifold near gear test apparatus.

(4) Temperature of oil going to each of the pedestal bearings of test unit.

(5) Temperature of oil from each of the pedestal bearings of test unit.

(6) Temperature of oil from test machine.

(7) Temperature of oil from drive gear bearings.

(8) Pressure of oil from lubricating oil coolers.

(9) Pressure of oil at manifold on test apparatus.

(10) Pressure of oil to bearings of test gear unit.

(11) Pressure of oil to drive gear unit.

The second system to supply oil to the test gears only shall have adequate provision to thermostatically control temperatures of the oil between 35° and 200° Fahrenheit. Sufficient remote reading gages and thermometers will be furnished to measure the following:

(1) Temperature of oil into heat exchanger.

(2) Temperature of oil from heat exchanger.

(3) Temperature of oil at nozzles.

(4) Pressure or vacuum within test gear case.

(5) Pressure of oil supply to test gear teeth.

The rate of lubricating oil flow to the test gear teeth will be measured by a flow meter. Nozzles are provided, to supply oil at top of gear mesh, or at the bottom. These nozzles can be rotated through 180° while unit is in operation and provision has been made so nozzles can be easily replaced. Nozzles with various sized openings will be provided, and thus, amount of oil and pressure can be varied.

k. Instrumentation

All instrumentation for noise readings and recordings, as well as arrangements for high speed photography work will be done by the New York Naval Shipyard Material Laboratory after installation of the apparatus.

TEST PROCEDURES

At the present time the test plan is to operate the apparatus at speeds above which noise and vibration data are to be taken. The desired torque will then be applied between the test gears and the drive gears; then the electric motor will be cut out and stopped quickly with the brake. As the speed of the unit decreases, being gradually decelerated by the flywheel until the speed is reached, at which noise data is desired, recordings will be made of the noise and vibration characteristics of the test gear set. At the same time, high speed photographs will be taken of the tooth mesh. It is hoped a correlation of the noise data, and the high speed photographs can be made to determine visually the conditions contributing to noise generation.

To enable the study of conditions varying from too much oil in the mesh and gear case, to too little oil, and too much air, the apparatus was designed to run from one extreme, the case full of oil, to the other extreme, whereby the case could be under vacuum.

To study the affect of backlash, and front lash, gear sets with varying amounts of backlash will be tested. Then in a gear with a large backlash, through the torque applier, one set of gears can be rotated relative to the other so backlash can be varied from zero to the full allowance. With the large backlash gear set, it is hoped to be able to position the gears so no contact occurs and operate them for a test run in this condition.

TEST RESULTS TO DATE

During the design phase of this work, Falk Corporation was authorized to conduct a test on a set of gears completely submerged in oil. The gear set was approximately the same size as the test gears and the purpose of the test was to determine the power required to operate gears completely submerged in oil. This would help in determining the size of the motor required for the test apparatus.

The test equipment consisted basically of a 30 horsepower dynamometer, rated at 2200 RPM, coupled to a gear unit, with 15″ centers and 4″ face, spur gears. The lube system consisted of a motor driven positive displacement pump, a cooler, a filter and a lube oil storage tank. The torque required to drive the unit was measured by a 21″ radius arm and a weighing scale.

The unit was operated as follows:

(1) Inlet oil to the unit at 0 and 10 psig and inlet temperatures of 130° F and 150° F.

(2) Single gear only, at inlet pressure of 0 and 10 psig and inlet temperatures of 130° F and 150° F.

(3) Single gear operation with vacuum conditions in housing of 10, 15, 20 and 25″ of mercury.

(4) Both gears in mesh, with vacuum conditions inside housing of 4, 15, 20, 25 and 27″ of mercury.

(5) Opposite direction, both gears, vacuum of 20 and 25″ of mercury in housing.

From tests conducted, it appeared that the test gears completely submerged in oil could be operated only under vacuum conditions. Under any other conditions, the power required by the test becomes too great to permit selection of a motor to drive the setup over a speed range of 1000/3000 r.p.m. as required by the specification. See attached Figs. 3 to 12 for power curves.

Due to these findings, a motor was selected to meet only estimated power losses under other conditions.

FUTURE STEPS

Falk Corporation now has a contract to construct the apparatus and is completing its design stage. Construction will begin shortly for delivery in the fall of 1955. After delivery to the New York Naval Shipyard Material Laboratory, the test stand will be erected, instrumentation installed and testing begun.

CONCLUSIONS

It is felt that this apparatus can furnish much information as to basic causes of gear noises and the resultant minimizing or isolation of these noises. Further, by precise control of the gears to be tested, to insure only one variable per gear set, the affects of helix angle, amount of oil and amount of air, and backlash, on noise generation can be determined. It is hoped much can be learned as to best lubrication practices and possibly at comparatively small cost, various additive oils can be tested for load carrying abilities. As the testing program progresses, various other information may be gained by the substitution of gears of different materials. All in all, it is felt this program will provide much of what has been missing in some other test programs, a fundamental understanding of gear noise generation.

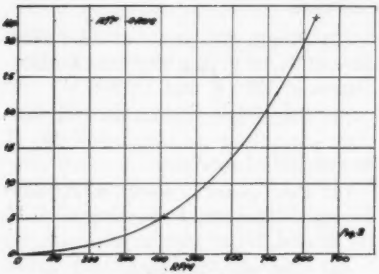
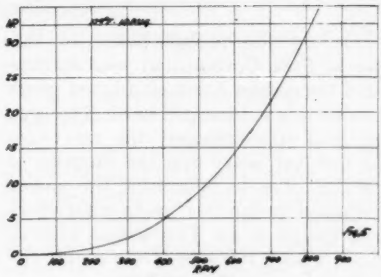


Figure 3



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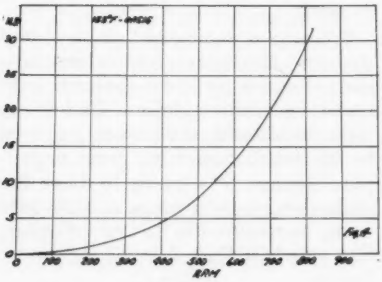

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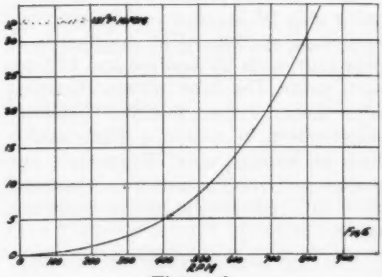

Figure 6

ACKNOWLEDGMENT

The writer acknowledges permission of the Bureau of Ships, Department of the Navy to publish this information. Assistance of the following has been of great help in the collection of material, and preparation and review of this paper.

Mr. K. R. Mercy, Materials Lab. New York Naval Shipyard.

Mr. R. J. Thomas, Falk Corporation.

Messrs. J. W. Sawyer, R. Bransby, H. J. Ball, J. D. Mayer, Bureau of Ships, Navy Department, Washington, D. C.

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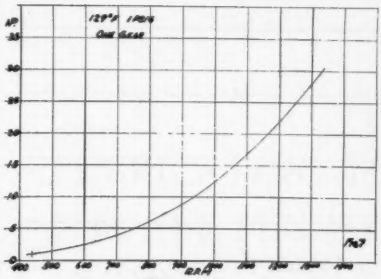

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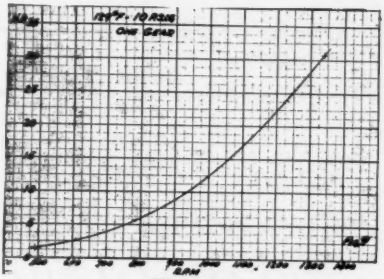
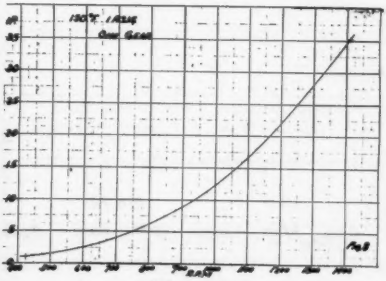


Figure 9



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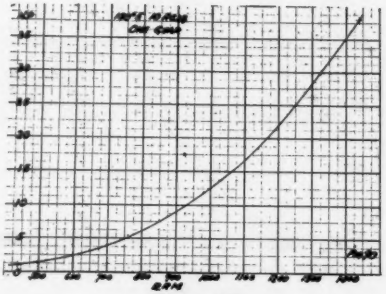

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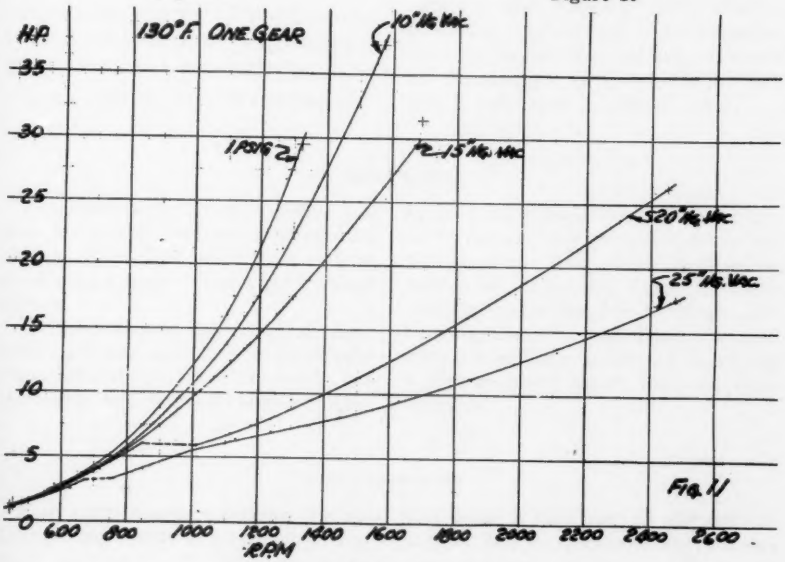

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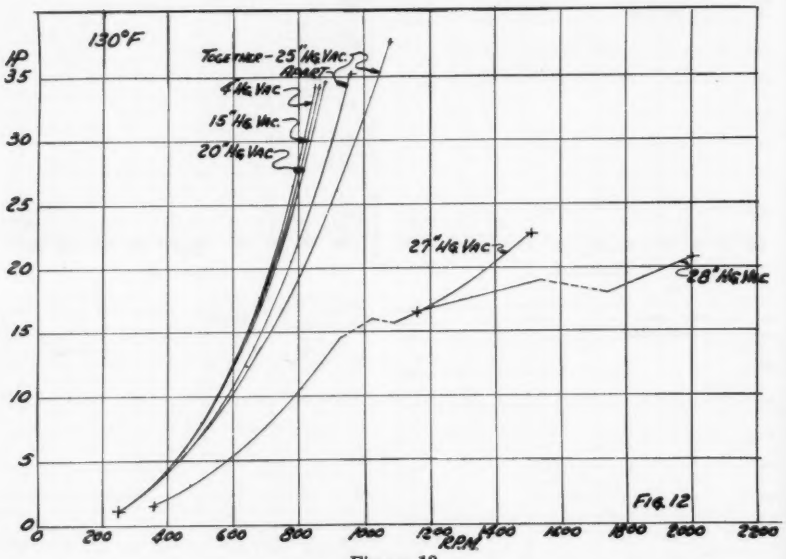
130°F
TOGETHER - 25" Hg VAC.
APART
4" Hg VAC.
15" Hg VAC.
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27" Hg VAC.
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1200
1400
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2000
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R.P.M.
Fig. 12

Figure 12

LUBRICATING TURBOJET ENGINES FOR ALL FLIGHT CONDITIONS—AT HIGHER AND HIGHER ALTITUDES

ACKNOWLEDGEMENT

This article was published in the March 1955 issue of *General Electric Review*. It was written by W. H. Wetmore.

Engineers agree that the jet engine's future relies strongly on advancements in aerodynamics, metallurgy, and machine design. Yet for all the emphasis in these directions, the importance of lubrication can't be overlooked.

Consequently, the lubricating system of a jet engine receives the utmost detailed attention. Engineers constantly chip away at the problem of lubricating — and keeping lubricated—a powerful high-speed high-temperature gas engine that not only operates at increasingly higher altitudes but also is subjected to high-G loads.

NO GARDEN HOSE

The whole objective of jet propulsion is to develop thrust from a momentum change. All design effort thus aims at producing a high-velocity jet exhaust stream.

If engineers could propel an airplane with a garden hose, they undoubtedly would. But they can't. They need a gas generator, which in a jet engine is considered everything forward of the exhaust nozzle. This generator needs rotating parts. And there begins the source of lubrication difficulties. For rotating parts require bearings, and bearings must be cooled and operated with a thin film of lubricant, known as a film-pressure layer, between their surfaces.

INNER WORKINGS

Four main bearings support the rotating components of a typical jet engine (Figure 1). To lubricate them, you need a tank for oil storage, oil pumps, jets to spray the oil in a desired pattern, oil coolers, filters, air

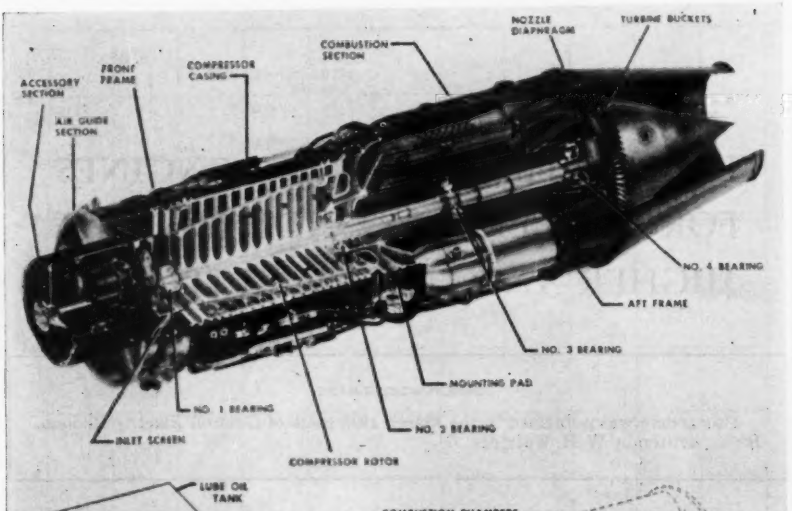

Figure 1. Jet Engine Lubrication System

vents, and all the associated oil lines.

Oil pumps are driven from a combination of gears, or accessory drives, powered by the compressor shaft. The forces exerted by these gears on one another are transmitted through films of oil between their contacting surfaces. Maintaining these film-pressure layers is still another function of the jet-engine lubrication system.

Additionally, the fuel-control system of many jet engines operates hydraulically. Under this condition the oil used as the controlling medium also lubricates the gears and bearings. And so as a third requirement, the lubricating system supplies sufficient oil to fuel regulators and other control components.

With these requirements in mind, let's look briefly at some specific functions of a lubricating system's components.

The tank for storing oil is normally mounted on the aircraft. Its capacity of 3.3 to 10 gallons is great enough so that oil consumption does not limit the range of an aircraft. Because the jet engine's lube system depends completely on gravity, the oil tank is located above the engine and high enough over the lube pump to insure a satisfactory pressure head.

The limited space in an aircraft requires that the oil tank be as small as possible. However, the oil is subjected to the churning action of bearings and gears and absorbs a lot of air, resulting in air entrainment and oil foaming. Thus the oil returned to the tank resembles a frothy milkshake. The tank must have sufficient storage capacity to decrease the air entrainment and oil foaming.

Absorbed air has a detrimental effect on the lubricating system because volumetric pumps are used to move the oil. In other words, an increase in air entrainment in effect decreases the amount of "solid oil" available for lubrication and cooling. Thus in practice you find fairly elaborate tank designs —some with built-in baffles, deaerators, or centrifuges—to assist in removing the air. The period of time that the oil remains in a given tank—known as the dwell time—is a function of the tank's size. And so to insure enough time for the entrained air to be removed—via vents, for example —the tank must be sufficiently large.

The volumetric pumps used to circulate the oil are not in themselves unusual, reliability being their most important feature. Several pumps are, however, incorporated in a single pump housing because the number of accessory drives are limited. (Accessory drives also power, for example, the aircraft's electric generator, tachometer, and fuel regulators.)

Normally you'll find three separate gear pumping elements, or sections, in a jet-engine lube pump. Two elements supply oil to the bearings, gears, and hydraulic fuel controls. The third section, the scavenge pump, draws oil from the bearing sumps for return to the storage tank.

Next is the matter of obtaining the necessary oil-spray patterns for the bearings. In practice this is accomplished by feeding the oil through converging nozzles, or oil jets, each so proportioned that the oil satisfactorily impinges on and penetrates the bearings at all engine speeds. But although the jet size must be small enough to adequately lubricate the bearing even at low engine speeds, it can't be so small that foreign particles in the oil will clog or block it. Frankly, the design is a compromise.

In one respect, lube systems are designed for the worst conditions that an aircraft can be operated under. Poor oil storage, plus the possibility of one aircraft's jet wake blowing dust into the inlet of another, could sufficiently contaminate the oil to cause bearing failures. For this reason, screen and magnetic filters are necessary to remove harmful particles. Generally, they will remove any foreign material down to about one fifteenth the size of a salt grain.

The most important single require-

ment of the scavenge pump is its ability to handle hot aerated oil at extremely low pump-inlet pressures. This function is, in fact, a direct measure of its basic capability. Then, too, the dive and climb attitudes of an aircraft may uncover one of the scavenge pump's inlet ports so that it becomes airbound. In short, it can't pump oil. To minimize this possibility, special attention is given to the design of the pump's internal passages.

HOT POTATOES

A jet engine is a hot potato; metal surfaces that contact oil are as hot as 500 F. Some provision must be made for cooling the oil to keep its temperature within reasonable limits. Utilized for this purpose is a heat exchanger that transfers heat from the lube oil to the engine's fuel.

The importance of cooling becomes more apparent when you consider that air entrainment and oil foaming are primarily functions of temperature. At extremely high temperatures any incompressible fluid bubbles and foams. The lubricating oil in a jet engine is an incompressible fluid; at elevated temperatures it is highly susceptible to trapping and entraining air.

There's an advantage to using the engine-fuel lube-oil-type heat exchanger: fuel entering the combustion chambers is preheated, helping it to atomize more readily and thus increase the jet engine's combustion efficiency.

Obviously, the best lubricating system—regardless of application—is one that can fulfill all its required functions with the least oil loss. In jet engines, most of this loss is in the form of vapor escaping from the engine. And so the lube-system engineer is fundamentally interested in recovering with a minimum of oil-vapor loss the oil that drains from the jet engine's bearings to the bearing sumps.

To minimize oil-vapor loss, a lower air pressure is maintained in the oil sump rather than in surrounding parts of the engine that the vapor might escape to. This way, air flows into the bearing sumps. (Oil-saturated air leaves the engine through special vents. Adequate vent control is provided to insure sufficient air-flow rate to maintain the low pressure requirement of the sump.)

Good air and oil seals around the bearings are essential to keep the proper pressure in the sumps. Oil seals perform the same function as the piston rings in your automobile engine. That is, they confine the oil in the compartment that it is discharged into. Air seals, on the other hand, prevent too high an air flow into the oil-bearing regions of the engine. Oil foaming and air entrainment are thus reduced by controlling the air flow into the oil-sump regions.

FLIES IN THE OINTMENT

Briefly, these are the components and functions of a typical jet-engine lubricating system. What types of troubles crop up in practice? Those confronting us today fall into three categories: those peculiar to the jet engine, the engine-aircraft installation, and the operational mission of the aircraft.

Plugged oil jets are one example of a lubrication problem pertinent to the engine. Here, foreign matter and the

deterioration of rubber components downstream of the oil filters are the primary causes of plugging. The best way to eliminate rubber deterioration is, of course, to improve the quality of rubber components. Elimination of foreign matter is less clear-cut, however, because it involves emphasizing to personnel at operational bases the need for better oil storage and oil-handling methods.

Oil coking is an ever-present problem in jet engines. During flight, oil is splashed against engine surfaces that may be as hot as 500 F. Decomposing at these temperatures, the oil forms on the engine walls a heavy carbon residue that may flake off and clog the oil filters, causing them to by-pass contaminated oil through the lube system. Fortunately, you can reduce oil coking materially by using insulation blankets to lower the bearing sumps' metal temperature. Also, baffles can prevent the oil from contacting hot surfaces.

Another problem peculiar to the engine is that of high oil consumption—the result of high temperatures, improper air pressures in the bearing sumps, and excessive seal clearances. Here the most effective countermeasure is to improve bearing-sump designs and air-pressure control.

Associated with the engine-aircraft installation is the difficulty of designing the oil tank. You'll recall that the tank must be as small as possible and consistent with the expected range and mission of the aircraft because of weight and space limitations. This in itself is easily done, but other factors complicate the task. The tank not only stores oil but also removes entrained air. Thus it must be large enough to insure a satisfactory dwell time. In addition, the tank's air vent must be carefully designed, because any improper functioning will result in a high loss of oil vapor.

Air pressure inside the aircraft fuselage in the region of the engine's air vents can cause trouble, too. If the fuselage pressure is lower than that in the engine's oil sumps, a high amount of oil vapor continually blows into the fuselage. To prevent this, ejectors are employed to keep air pressure in the oil sumps lower than that inside the plane's fuselage.

Lubricating problems peculiar to operational missions, the third category under consideration, generally result from aircraft maneuvers, such as climbing and diving. Three specific situations can arise that indicate malfunction or failure of the lube system.

As an example, a pilot might find that his engines are consuming a large amount of oil. Yet his oil-pressure gage will show no drop in pressure nor will his engine indicate an inability to maintain power. Under certain other flight conditions, an oil-pressure gage indicates an unusually high oil consumption, but there is no malfunction of the fuel-control system that is hydraulically actuated by lube oil. The third and most acute situation occurs when oil consumption is so high that oil-pressure loss prevents the necessary fuel-control components from functioning properly and seriously impairs the engine's ability to maintain power.

Sustained climb attitudes are another operational problem. Some interceptor aircraft may be in a climb attitude for 8 to 12 minutes—more than enough time for all the oil to drain into the rear sections of the engine. If the bearing seals and scavenge pump aren't functioning properly, oil pressure will be lost before the climb is completed and the pilot will have to discontinue his flight and return to base.

Fighter aircraft are characterized by their ability to maneuver rapidly in

climbs and dives. Also, a fighter must be versatile enough for use as a dive bomber when necessary. At such times, high-G pull-outs will force all the lube oil out of the engine's mid-frame air vents, materially reducing the fighter's combat capability.

Bombers, like fighters and interceptors, have their own particular lubricating problems. For example, occasionally in flight a pilot will have to shut down one of the bomber's engines because of fire warnings. Normally this shouldn't affect its tactical mission. It does mean, however, that the jet engine will be windmilling possibly for several hours. During this period the lube oil supply might be shut off; thus the oil on the bearings at the time the engine is shut down is the only lubrication available.

On certain heavy bombers, auxiliary jet engines are used for take-offs and climbing but not for normal aircraft cruising. Thus during most of the flight these engines windmill. But even with their lube system functioning, oil from the jets impinging on the bearings is inadequate. For the engines windmill at such low speed that the oil pumps—operated through the accessory drives—can't build up sufficient oil pressure for the proper spray patterns.

These, then, are a sample of the lubrication problems in a typical jet engine. Despite them, the jet engine of today is a remarkably reliable power plant, with its lubrication system doing a good job. Last year, for instance, a B47 *Stratojet* bomber powered by six jet engines completed the equivalent of 10 trips around the world without requiring any engine overhaul.

LOOKING AHEAD

You can be sure that new lubricants and lubrication systems will play a vital role in the future of the jet engine. They will permit operation under increased bearing loads through a greater range of ambient temperatures and at altitudes not now envisioned.

These systems will probably employ synthetic oils rather than the natural petroleum oil used today. For one thing, present lubricants operating at existing engine temperatures begin to boil at 40,000 feet and bubble badly at 50,000 feet. High oil loss results from foaming and evaporation. Synthetic oils appear to be a solution to this problem.

Vertical take-off aircraft will alone present many new lubricating problems. The extended time that the engine is in vertical position requires better scavenge pumps and seals than present types; new lubrication-system designs will be needed.

Among the lubricating engineer's long-range goals, one stands out above all—the development of a lubrication system as effective and functional as that in the human body. Consider the number of times you flex a finger, bend an arm, or move a leg. Think of all the other forms of human motion that require lubrication.

Our human lubricating system is the most amazing and ingenious of all. It is the one that engineers should best understand to find some new approach in the mechanical design of newer and better jet engines.

MAKING ATOMIC PILES BEHAVE

ACKNOWLEDGEMENT

The author of this article is Ivan M. A. Garcia, an electrical engineer at the Hanford Atomic Products Operation, Richland, Washington. The article appeared in the March 1955 edition of *General Electric Review*.

Nuclear reactors represent an investment of millions of dollars that the engineer must protect with properly designed safety circuits and devices. A properly controlled reactor is a useful device, but if the safety circuits fail, it can become a potential package of uncontrolled energy.

REACTOR CONTROLS

A nuclear reactor can be controlled by inserting a neutron-absorbing material, by removing charges of reacting material, or by changing the physical interrelationship of the critical materials.

Proper positioning of a highly neutron-absorbent media, usually rods, is a common method for maintaining power level. When the rod is removed, more neutrons are available and the power of the reactor increases; wnen inserted, fewer neutrons are available and reactor power decreases (Figure 1.).

The rods can be actuated by electric or pneumatic motors through geared drives and controlled from a central desk. For quick shutdowns, safety rods can be inserted rapidly into the reactor, triggered by signals given by sensing circuits or at the will of the control-desk operator. To operate even during a complete power failure, the rods must use some sort of stored energy for motive power.

For control-rod drives, the engineer has applied various industrial systems: two a-c motors with differential gears and reversing switches; adjustable-speed a-c motors of the brush-shifting type; a-c gear-motors with reversing switches; d-c motors with field control and fixed armature current; and d-c motors with armature control and fixed fields. Ingenious combinations of these systems are being used successfully (Figure 2.).

For the safety-rod drives, winches with magnetically held clutches are used.

A typical control-rod circut (Figure 3) uses a feedback closed-loop system for position control. The oper-

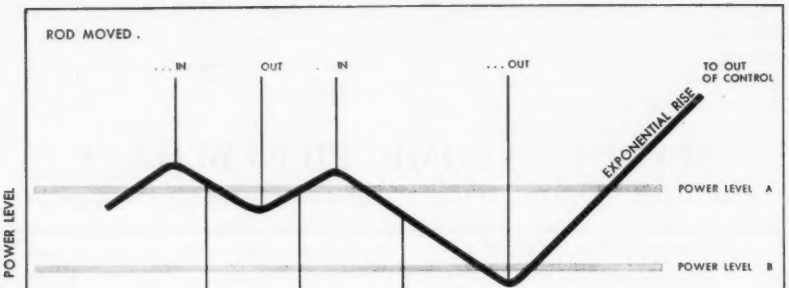

Figure 1. Nuclear Reactors can be controlled by inserting rod-like neutron-absorbing materials. In this method, removal of the rod results in more neutrons and increased reactor power; insertion of the rod causes fewer neutrons and, consequently, decreased power.

ator at the control desk turns a switch handle clockwise or counter-clockwise; the rod moves in or out of the reactor, depending on the direction of the switch. Speed of response is proportional to the amount of the switch handle's displacement, and the length of travel is proportional to the length of time that the switch is displaced. Limit switches control the travel in both directions, with lights indicating when the limits are reached.

Auxiliary devices show the operator the rod location at any particular instant, and brakes prevent creeping of the rod. The system must operate positively and efficiently both in magnitude and direction.

The circuit consists of a number of sentinels that monitor certain conditions and properties, reporting through time-delay relays to a series circuit that controls a set of trip relays. Powered by the relays, magnetic clutches hold the rod that, accelerated by gravity, falls into the reactor when power is cut off. Again, limit switches control the length of travel, with lights indicating whether the rod has penetrated. Annunciator drops single out the abnormal condition and hold a target and an alarm until acknowledged. By-pass switches enable different portions of the circuit to be worked on or removed from the system. Time-delay relays are adjustable to set up a system of priorities in the operation of the various devices, the most critical taking the highest priority.

The engineer finds little difficulty in applying known and tested types of industrial adjustable- and variable-speed drive systems to the essentially similar problems of reactor rod control. Sometimes, the engineer finds the components readily available from leading manufacturers; at other

times, the parts are fabricated. For the latter purpose, industry has always met or surpassed the required characteristics.

SENSING CIRCUITS

Entrusted with the most important job in a nuclear-energy installation are the devices in the circuit, Figure 3, that monitor the critical conditions and properties of a reactor. This applies to the experimental research reactors and the production reactors at AEC's GE-operated Hanford Atomic Products Operation in Richland, Wash., and will also be true of power reactors. For best efficiency, these machines must operate at the highest neutron fluxes and the highest temperatures consistent with safety both to reactor materials and operating personnel. If certain limits are exceeded, disasters can occur that render the reactor completely unusable and ruined for posterity; further, contaminated products could be spread to a large area. Because reactors are one of the most expensive devices ever built, the utmost care must be taken to prevent exceeding the limits established for efficient operation.

Monitoring neutron flux—most important of the characteristics—is accomplished by ionization chambers. Strategically located throughout the reactor, ion output is amplified by electronic amplifiers, with the resulting signals fed to the safety circuit. This output is proportional to the neutron field intensity at the point where the chambers are located. The physicists have provided the relationships necessary to correlate such readings with the total, maximum, and average flux in the different regions of the reactor. Electrically, the signal resolves itself into a "go" or "no go" proposition to continue operation or to shut down. And when a shutdown, or "scram" signal is received, we wish we had shut down already. Scram is a colloquial term for a quick, unexpected shutdown of a reactor.

The Hanford reactors use large amounts of water as a coolant to remove the heat generated during operation. Other reactors use air for their coolant, and the power reactors now in the drafting-board stage may use liquid metals as a heat-transfer media for the production of useful power. As it enters the reactor, the pressure of this closely regulated coolant is monitored by common industrial pressure switches installed at the inlet headers. Again electrically, the circuit sees either a closed or an open contact at the switch.

The coolant's exit temperature limits the power level of reactor operation. If allowed to rise indiscriminately, a liquid coolant can flash to vapor with subsequent damage to the reactor and fuel charges. Through relays, thermocouples give the "go" or "no go" signal that is also fed into the safety circuits.

An earthquake can cause lateral or vertical displacement of the reactor materials, preventing the rod from penetrating into the active zone. Thus at the first indication of a heavy quake, a seismoscope triggers the safety circuits through a relay, with the signal again appearing as a closed or open contact on a relay.

Loss of power to a reactor building, its auxiliaries, and controls is the equivalent of a car hurtling down a highway at an ever-increasing speed, the driver having no control of steering wheel or brakes. Destruction would be certain. Undervoltage relays of proved industrial types watch

the voltage and signal when it falls below a prescribed limit. These undervoltage relays monitor the voltage at the main substation, the building incoming lines, and the feeder to various rod controls. Through a power-failure relay, they also indicate whether operation should be continued or shut down.

When the operator finds its necessary to quickly shut down the reactor, he depresses a push button at the control desk. Then the circuit sees electrically a pair of contacts open or closed.

Sometimes a multiplicity of these sensing devices is used; at other times they are used singly. Before a reactor can be brought up to power or even started, the safety circuit must see each one of the devices at its proper value.

INTERLOCKS

Interlocked circuits to exclude all possibility of human error or misoperation are desirable but, of course, impossible. The unrestricted addition of interlocks quickly makes the whole system of controls inoperable, forcing the engineer to strike a balance. He must remove from human control potential hazard possibilities that would result in disaster or operation without safety devices and yet leave the circuits flexible enough to protect in all other situations. The bypassed interlock is worthless; the trigger-happy one, an operational nuisance. Judicious use of interlocking is a must in a good design job and an asset to the continued safe operation of a nuclear reactor.

A typical interlock circuit has, at different points, four ionization chambers that monitor pile activity. Each chamber can be bypassed, provided an alarm indicates that the respective safety device is no longer in the circuit. If two chambers are removed from the circuit and an attempt is made to bypass the third one, annunciators, or alarms, are no longer reliable and the reactor must shut down automatically.

Interlocks protect equipment, limit travel, restrict operations to orderly procedures, and protect areas and personnel. Throughout the circuiting, the engineer applies them in the form of key-operated and limit switches, interposing and timing relays, and even mechanical linkages.

Not considered a serious design problem except when carried to excess, interlocking in this application is in many ways simpler than in a continuous process industrial plant.

BYPASSES

Although bypasses appear to be in direct opposition to the use of interlocking, they are necessary to enable hot maintenance of the safety circuits. "Hot" means that the series safety circuit is energized through the bypass switch and the sensing device circuit de-energized for maintenance or other reasons. Also, with experience in reactor operation, a particular monitor could be judged not so critical and its bypass switch left closed. Bypasses also enable certain sensing devices to be in the circuit for short periods of time, such as during start-up of the reactor, and then removed for the continued operation at higher power levels.

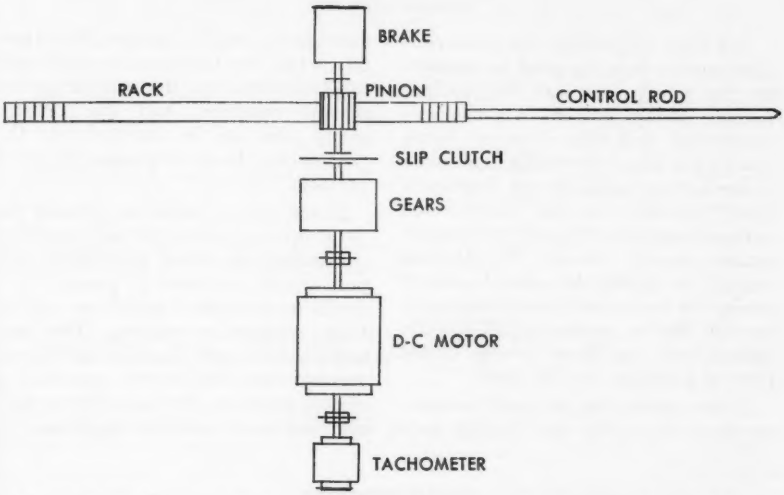

Figure 2. Engineers applied various industrial systems for the operation of the control-rod drives, and ingenious combinations of these systems are now being used successfully.

The mechanical features of the typical bypass control switch include a target for visual indication, a key-operated lock in the handle or above the switch, and spring return to neutral, or normal, position, known as "unbypassed"—a term meaning a bypass switch in its normal position.

INTELLIGENCE

Target-drop and bull's-eye annunciators keep the main operator in the control room informed of the vital circuits' status throughout the building. This large scoreboard tells at a glance what equipment or circuit has misoperated, the drops being arranged in the relative order of their importance. Industrial-type annunciators without modifications have been used in this link of communications. However, with a large number of drops, the size becomes too large and takes vital space off the control panels. Need is great for a simple annunciator with a memory of at least three drops.

A typical annunciator circuit operates like this: When the remote contact closes, it energizes a relay, lights a bull's-eye, and rings a bell. The operator silences the bell by operating the switch. When the fault is corrected, the light goes out and the bell sounds again, reminding the operator to reset his silencing switch.

Other visual indications in the form of meters, counters, lights, and recorders supplement the main annunciator in delivering information to the control and monitoring room. The circuits for these are simple, usually consisting of a normally open remote contact in series with an indicating light.

POWER SUPPLIES

The force of gravity—the most reliable known—can be used to accelerate the safety rods into any nuclear reactor. Always present, it is also unidirectional, and this direction being downward does not readily lend itself to use for horizontal travel. Engineers have therefore devised complicated hydraulic systems (Figure 4) with potential energy stored in elevated masses to enable the use of gravity power for horizontal penetration of a reactor. But an equivalent all-electric system with the stored energy in the form of batteries can be used.

Utility power can be used to operate both the safety and control rods through a-c and d-c motors. But when this is lost, the horizontal control rods can be driven into the reactor by the storage batteries, and the vertical safety rods can be inserted into the reactor by the ever-present force of gravity.

Great care is taken to protect the batteries from electrical and mechanical damage, and their capacity in ampere-hours is chosen to permit hundreds of successive shutdown operations without recharging. The batteries can be used also for station control batteries, the electric operation of circuit breakers, the annunciator supply, and control-circuit supplies.

DESIGN PRINCIPLES

In the design of control circuits, the engineer draws freely on the experience of industrial application engineers, as well as that gained from the operation of reactor controls for nearly a decade. Although no hard and fast rules can be given for circuit designs, certain principles have slowly emerged and are evident in job after job. These furnish a rudimentary check list for measuring a design's fundamental correctness. Several principles follow.

Duality

Every important utilization point has two complete dual channels of power feed—normal and alternate. This is carried through at the main building substation with two incoming lines and two transformers, duplicate feeders to motor control centers, and duplicate feeders to lighting panels. Great efforts are made to electrically and physically isolate the parallel feeds so that failure in one will not affect the other.

For each critical control function, two relays are used with one backing up the other. Their coils are in parallel, with interlocks connected to a common reset button.

Fail-safe

Because every reactor control circuit must be fail-safe, the engineer must anticipate misoperation resulting from personnel errors. He must further guard against it by so interlocking the circuits that, at worst, human failure results in an accidental shutdown of the reactor without permanent damage to any of its vital parts. Even deliberate attempts to disable the reactors by malicious misuse of control components must not be successful.

Similarly, to a larger degree, equipment failures must be guarded against. Circuits are always so designed that the reactor will remain under control or will shut down at the time of the failure. Worst offenders in this respect are holding relays, burned-out coils on continuously energized circuits, and sticking relays.

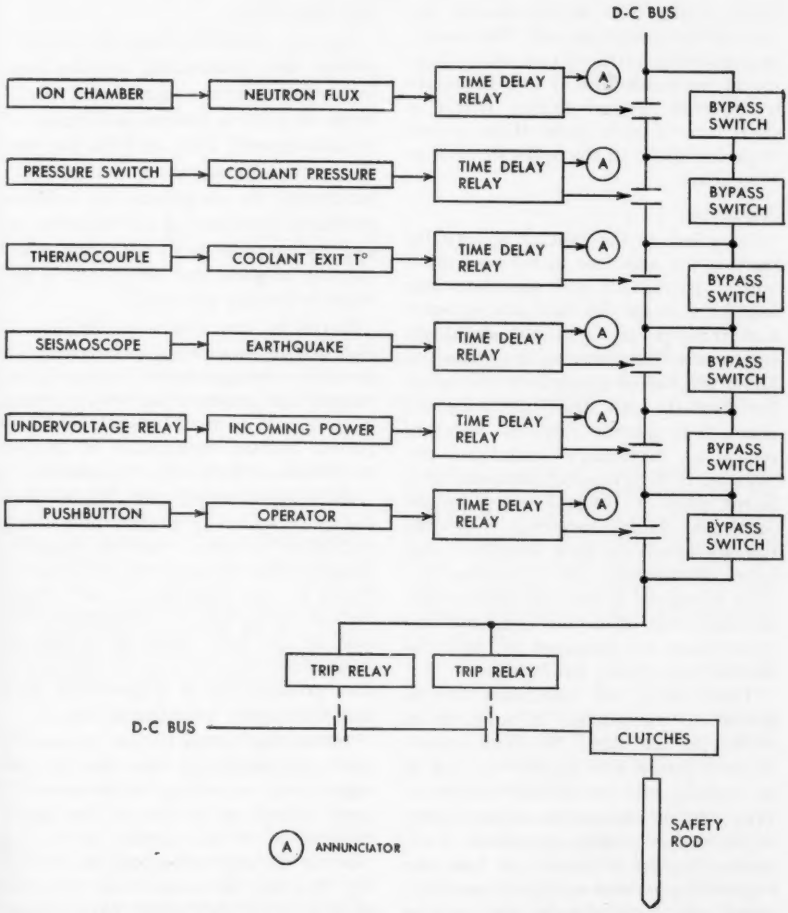

Figure 3. A Control-Rod Circuit consists of a number of sentinels that monitor certain conditions and properties, reporting to a series circuit that controls a set of trip relays.

Examples of fail-safe operations are loss of a pressure switch seen by the circuit as low pressure; loss of an undervoltage relay seen as power failure; and accidental jarring of a seismoscope interpreted as an earthquake. In other operations a duplicate facility takes over the operation; if the time interval permits, automatic transfer of one to the other is effected, with proper annunciators warning the operator that he is on the stand-by device.

The engineer must also foresee misoperation caused by a series of unre-

lated events that would disable the control or safety circuits. He evaluates the probability of two relays failing at the same time or two physically isolated devices failing within a short time of each other. He may talk of probabilities in the order of once in 4000 years.

Sequential

Long before the reactor is actually started, the engineer in his own mind and in discussions with associates has taken it through the start-up sequence and through innumerable shutdown sequences. The circuitry is checked to allow the varied operations necessary for start-up, and the interlocks are checked to permit only one order of operation. Necessary variations are covered in a bypassing sequence that forms part of the operating procedures. A similar sequence organizes the operations for slow shutdown. For quick shutdowns, the time-delay relay's sequence is checked and co-ordinated with other circuits, and bar time charts are prepared for all foreseeable emergency conditions.

Total times are estimated for all shutdown sequences in number of cycles (60 cps base). Whether normal or emergency, the shutdowns are in an orderly and predictable sequence. The random sequence—neither predictable nor easily provided for—poses the real problem. At best, the engineer provides a tightly co-ordinated set of circuits for the various systems.

And Reliability

Circuit reliability and circuit simplicity are practically synonymous terms for reactor controls and auxiliaries. A control scheme is divested of all unnecessary frills, and the number of circuit components is reduced to a minimum. In weighing the relative merits of two control schemes for an application, the one with the least number of parts and nonstandard devices is usually selected.

Equal in importance to circuit reliability is equipment reliability. To develop a design created around load-center unit substations, motor control centers, and control cabinets, the engineer obtains equipment of proved performance from reliable vendors.

Although a vague and elusive term not measurable in dollars and cents, reliability becomes a strong argument when evaluated in terms of shutdown hours or the duration of the outage caused by its failure. Frequently, the cost of the best piece of equipment more than compensates for the cost of lost productivity if it prevents even one shutdown in its design life.

Reliability is not limited to circuits and equipment: it can also be obtained by providing such items as spare relays in a cabinet and spare feeders in a duct bank; by having sleeves through shielding walls; and by choosing manufacturers for their ability to furnish spare parts, repair facilities, and service engineering.

PRACTICAL DESIGN LIMITS

Many a concept generally assumed to be true is shattered when the behavior of a physical system does not follow the theoretical norms that it was designed for. Such discrepancies arising mainly from mechanical, magnetic, or electrical inertia are usually of no great significance in everyday applications. But they must be recognized and allowances made in the design of reactor control circuits.

It might be assumed that the force of gravity will accelerate the safety rods at the rate of 32 feet per second

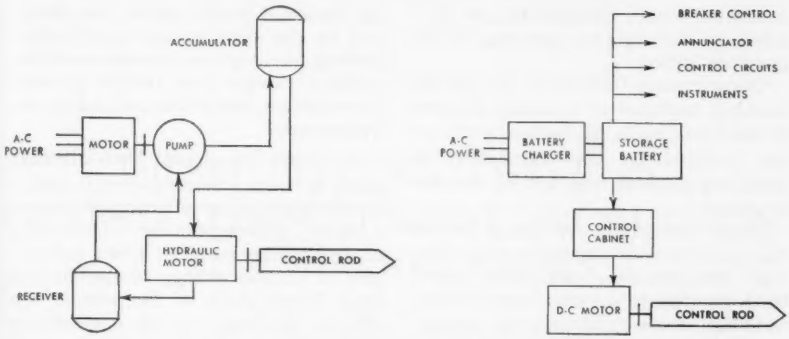

Figure 4. Horizonal penetration of a nuclear reactor can be accomplished by one of two systems, both utilizing the force of gravity: the hydraulic system with energy stored in elevated masses and the electric system with energy stored in the form of batteries.

per second into a reactor. However, they are not free bodies because they must accelerate the drums and overcome the drag of the winch cables. This increases not only the time of travel but also the time in which a sufficient length of rod is inserted into the reactor to check the nuclear reaction. In the short times considered, this appears as a large percentage.

It might be assumed that as soon as the safety relays open their contacts the rods begin to fall. But with continued operation, some residual magnetism will be present in the clutches, thus introducing a measurable time interval before they release the rods. This time interval varies from clutch to clutch and increases with the time of operation.

Assume further that by closing a switch a relay will pick up instantaneously. However, the magnetic inertia of the coil, plus the mechanical inertia of the armature, will provide enough time lag to appreciably affect some of the total times calculated for the bar charts.

The designer is also concerned with practical limits of equipment. Often, he finds his circuits curtailed by the number of contacts available in a relay or by the maximum capacity of items of standard manufacture. At other times the quantity is not economical for the manufacturer to make special runs.

If the practical limits of delivery cannot be met and the production capacity of manufacturers is unavailable, substitutes have to be accepted and new designs developed for the obtainable equipment.

The engineer is limited because characteristics and appearance of extensions to already installed equipment must match and line up with the existing equipment. Plant standards; availability of spare parts; and mechanical, thermal, and nuclear properties of materials are other limiting factors. Sometimes the answers to such items as the effects of radiation fields on insulating materials are not fully known.

Certain space limitations restrict the amount of cubic inches for circuit components—especially in the faces of

a reactor where dimensions are dictated by the rigorous spacing of the graphite lattice.

Other critical limitations are on the number and size of conduits allowed in shielding walls. Radiation windows are avoided by the calculation of shielding equivalences and by the use of offsets.

These conditions are but a few of the variables of control-circuit design. With the exception of the problems peculiar to nuclear reactors and radiation, these conditions are similar to those encountered by engineers in other industrial endeavors. By the judicious use of the number and types of a system's components, by the careful choice of conditions to shut down, and by the discriminate use of interlocking, the engineer creates a simple, reliable design that fulfills process, construction, operation, and safety requirements.

But what lies ahead? Two ultimate goals in reactor control-circuit design are the production of a completely automatic self-regulating closed-loop control system and the direct conversion of nuclear energy to electric energy. These goals of the atomic age offer a challenge to the engineering professions. And as leaders in the production of power sources, the engineers will be expected to meet this challenge.

DIESEL ELECTRIC DRIVE VS. GEARED STEAM TURBINE PROPULSION FOR SUPERTANKERS

T. ORCHARD LISLE

THE AUTHOR

is Editorial Director, *The Oil Forum*. His engineering career began in 1901. In 1908 he became assistant editor of the *Motor Boat* in London, and has been identified with trade and technical journalism ever since. During World War I he was a member of the Navy's anti-submarine board. He was unable to accept a U. S. Navy commission in 1916 because his naturalization (Mr. Lisle was born in England) had not become final. In commenting on this in 1919, the Assistant Secretary of the Navy, Franklin D. Roosevelt, wrote: "I feel certain, however, that you can feel satisfied that you have rendered excellent service to the cause through the medium of *Motorship*."

Mr. Lisle's editorials and associated efforts between the wars, during World War II and since, have contributed positively to the resolution of many problems of an international and national nature.

The present article is a talk which was presented before the American Institute of Electrical Engineers on 14 March 1955.

While there definitely has been a surge towards steam power for supertankers this past two years, there were under construction in the free world shipyards at the end of 1954, 199 Diesel tankers aggregating 1,205,579 gross tons, compared with 114 steam tankers totalling 1,892,647 gross tons. With other types of large ships building the Diesel engine still predominates.

The subject of this paper is quite controversial, being a comparison of Diesel-electric A.C. drive with geared steam-turbine propulsion for supertankers, with particular reference to ships of about 32,000 tons deadweight capacity and 12,500 s.hp. Perhaps I am foolish to give my views before a group of distinguished engineers, especially as angels may fear to tread on the engine-room gratings. But, I am hoping that the electrical fraternity in this audience will have a sympathetic understanding, even if dyed-in-the-wool steam men may show no mercy should they find weak points in my statements.

Unfortunately, there are no Diesel-electric propelled supertankers in service to provide authentic data. In fact, there is only one large commercial tanker in the world fitted with ma-

chinery of this type. I refer to a 16,000-ton 4,800 b.hp. Shell oil-carrier that has been operating for several years. She has had one of her four Diesel engines pulled out, and replaced with a gas turbine as an experiment. This combination machinery is to be replaced shortly by a 6,500 s.hp. gas turbine for development purposes. Several years ago the U.S. Navy built 23 Diesel-electric tankers of about 4,000 tons dwc and 6,500 s.hp, some of which are operated by private concerns for the M.S.T.S.

During World War I the late Charles Schwab, acting on behalf of the U.S. Government's Emergency Fleet Corporation, ordered eight Diesel-electric tankers from the Sun Shipbuilding & Dry Dock Company, but the Armistice caused a cancellation. These tankers were to have a number of small engine-rooms on deck, and only a propelling motor-room down below.

We have to remember that the speed of a tankship is variable, being governed by weather conditions, and by the extent of ballasting the hull when no cargo is aboard. On some stormy voyages she could have her speed cut-down and fluctuate from around eight or twelve knots to full speed for varying periods. When the power of a geared-turbine tanker is lowered, and her speed reduced, the fuel consumption per s.hp. hour goes up, which is not the case with a Diesel-electric vessel, as one or two generator units can be shut down temporarily and the others maintained at full power.

Therefore, in making comparison, it is necessary to assume that both ships are operating continuously at full power, and this method has been adopted in this paper. All figures in this paper are necessarily hypothetical in part. However, they should be close enough to facts to be reasonably accurate.

Diesel engines, generators and propulsion motors have been manufactured and installed in naval and coast guard vessels of 12,000 s.hp and up, such as mother ships and icebreakers, where they have given an excellent account of themselves. Now under construction in this country is a Coast Guard icebreaker which is being equipped with Diesel-electric propulsion machinery of 24,000 b.hp. consisting of ten 2,400 b.hp. generator units. With 7% loss in transmission this means 22,320 hp. at the propeller shaft, but I understand she will be at 20,000 s.hp.

The reason why there are no Diesel-electric supertankers in service under the American flag is that oil company marine department officials either have been unwilling to "stick their necks out" sufficiently to build such a vessel and thus provide proof of the merits, or demerits, of this form of power, or they have not yet been convinced as to the economic benefits to be gained. These marine officials know from long experience what results they can obtain with geared-turbine drive, and evidently consider that they are on firm technical ground, so why risk a costly "experiment"? A modern supertanker costs $6,000,000 more or less, and an engineering mistake could well be very expensive and perhaps even cause them to lose their jobs. I can understand this attitude, but I am not sympathetic with their reluctance because it is their duty to encourage the engineering advance of any promising machinery.

The oil industry as a whole is very progressive, but as I view it, the marine departments have not made the same bold and far-sighted progress as have the divisions responsible for the development of drilling, production and refining. Allow me to take two specific instances. The Ohio Oil Company decided to search for oil at a depth hitherto unproven to be petroleum bearing. A special Diesel-powered rig was used, and a hole was made

to a depth of 21,482 ft., or more than four miles down. But, the structure they hoped to pierce apparently is below the expected level—at least at that particular location in the Paloma field in California. They got a stuck drill-pipe and a dry hole! After spending $2,225,000 in 38 months, the well was abandoned. They now know that it is possible to drill to this depth and I'm sure they will have another try. It was a calculated risk.

The Creole Petroleum Corporation has very recently completed a giant gas pressure plant on Lake Maracaibo in Venezuela at a cost of $20,000,000, and which will give an increase of 50% in output of existing oil wells. The power units consist of ten gas turbines of 6,000 hp. each driving rotary compressors, which force gas into the subsoil of the lake at 1,935 psi. This is a far greater pressure than ever before attempted with this type of compressor. In fact, it is double the pressure! It is also the largest installation of gas turbines. Bold vision backed by confidence was an integral part of the decision to spend $20,000,000 on this project. The next plant is to be twice the size.

Taking all indications to date into consideration, I can't quite see any American oil company marine department boldly ordering to-day, say, three 6,000 hp gas turbines for an electric drive supertanker. You gentlemen can be the judge whether the reason is the result of doubt, fear or perhaps plain common sense.

What is the result of the reluctance shown by American oil company marine departments? Definitely, such reluctance is not in the interest of engineering and economic progress, so perhaps it is time for oil company directors and coordinators to arrange for independent studies and thus obtain unbiased facts.

Steam tankers of to-day show extremely little gain in propulsive economy over tankers of twenty years ago. The comparative small saving in fuel consumption that has been made during the past few years has only been gained at the cost of raising boiler pressures and temperatures to a point where special and costly alloy steels are essential, and where the danger to engine-room personnel could be great if a feed line burst.

In the case of Atlantic Refining's supertankers using steam of 625 psi and 1,002 deg. F., the fuel consumption was reduced to about 0.5 lb per hp hour. Frankly, I would not like to be in their engine-rooms during wartime. If temperatures and pressures were raised any higher, it is not likely that anyone in the machinery compartment would escape serious injury or death should a piece of shrapnel hit a steam pipe. With the threat of war continuously hanging over our heads, this might give oil companies something to think about.

In the United States, steam pressures and temperatures of geared turbine tankers seem to have been established at around the 600 psi, 850 deg. F. level, but with some variations. So we can hardly expect much improvement in fuel consumption, or even common attainment of the 0.5 lb. per s.hp. of the Atlantic Refining tankers. Cities Service's new supertankers have a steam pressure of 850 psi, at 850 deg. F. In any event any gain will only be slight, and mainly with ships of 20,000 s.hp. and over. As a matter of fact, the consumption of all steam tankers has a tendency to increase after the first few years of service. This is not the case with Diesel engines, as their fuel consumption is more constant over the years.

With steam at 450 psi and 750 deg. F. a fuel consumption of 0.60 lb. per s.hp hour, and over, is common. Hence, I am being very fair in using a figure of 0.55 lb. for comparison purposes.

The highest steam pressure and temperature ever used for ship propul-

sion is that of the U.S. destroyer *Timmerman,* namely, 2,000 psi and 1,050 deg. on one shaft, and 875 psi and 1,050 deg. on the other shaft. Heaven help the engine-room crew if a high-pressure pipe lets go. With one larger warship the steam pressure is 1,200 psi with a temperature of 975 deg. F.

To go above 950 deg. F steam temperature it is necessary to use 2½% chrome molyalloy for steam piping, and austenitic alloy for part of the superheater. In fact, about 9% chromium is needed for the last pass of the superheater when steam is that high, as well as use of heavier drums and piping. These things add to the first cost of the plant. However, there is an overall weight saving when very high pressures and high temperatures are used.

Since the time I was asked to prepare this paper, the Maritime Administration has decided at the recommendation of Congressional leaders, that any bulk-oil carriers built under a Government-sponsored program, such as the trade-in and build plan, shall be capable of a speed of 18 knots.

This ruling will increase the economic value of Diesel-electric drive. All that will be necessary to obtain the higher speed is the installation of two or three additional Diesel-generator units, possibly on a flat in the engine-room, and have a more powerful propulsion motor. The total shaft horsepower required probably would be between 18,000 and 20,000, but that is for a naval architect to decide. The propeller would have to be designed to give good efficiency at either 12,500 or 20,000s.hp, so would be a compromise.

The extra power units need only be operated in emergencies or, for instance, to enable a tanker to reach port before quarantine closing time, and so not have to anchor overnight. Under normal voyage conditions the spare power-units could be idle, and the rest of the machinery run at its most efficient load. Whereas, with a steam tanker of 19,000 s.hp the fuel efficiency would be lowered when operating continuously at the reduced load of 12,500 s.hp, and she probably would burn 90 tons per day, or at the rate of 0.7 lb. per s.hp hour. When operating at 19,000 s.hp. she would burn 112 tons per day.

As the fuel efficiency of today's marine steam plants increases with higher powers, the fuel consumption is likely to drop to 0.55 lb. per s.hp hour at 19,000 s.hp, which would mean 111 tons per day, at 18 knots speed. The Diesel-electric job would have a consumption of approximately 92 tons a day under the same conditions, or 60 tons at the reduced speed.

During the next 15 years, or roughly the economic life of a tanker, many trade conditions could arise when it would not pay to operate a 32,000-ton tanker at 18-knots, and an owner might well be saddled with an inefficient geared-turbine carrier. He cannot shut down a part of the propulsion plant.

The greatest progress in steam-turbine tanker design during recent years has been larger and faster vessels with more rapid cargo-discharging facilities, better quarters for crew, and more efficient navigation equipment. This reminds me that some years ago your author proposed 30,000-ton deadweight tankers for Persian Gulf service, and the reaction was as "enthusiastic" as the current attitude towards Diesel-electric drive.

Diesel engines of a power, revolution-speed and light weight suitable for driving generators for electric transmission to the propeller are now manufactured by six American engineering companies of repute. They are, General Motors, Cleveland Diesel Division; Cooper-Bessemer; Nordberg Manufacturing; Fairbanks-Morse; Baldwin-Lima-Hamilton; and Enterprise Engine Division of General

Metals Corp. If engines of 1,000 rpm. and over are acceptable, the G.M.C.-Detroit Division can be included. When coupled to generators, any number of units can be installed to supply current to an electric propulsion motor of high enough power to propel the biggest and fastest tankship planned today. I already mentioned that the highest-powered American marine Diesel-electric propulsion plant now on order in the U.S.A. is 22,230 s.hp. Regrettably, this is not for a tanker!

Possibly, another reason why there are no American-flag Diesel-electric tankers in service is that the engine manufacturers do not appear to have pushed their wares very hard with oil companies. I know that in some cases they were disappointed to find themselves up against a brick wall, so became quite discouraged.

Just how far is it practical to go with Diesel-electric power? Studies have been made by one American oil company of 16½-knot tankers of 1,000 ft., 37,000 s.hp length, and 700,000 bbl. cargo capacity on a draft of 44 ft. to transport crude oil with full bunkers for a 25,000 mile round voyage but nothing has been done about it. The studies show that oil could be delivered with a transportation cost of less than crude can now be moved across Arabia by pipeline and carried in existing steam-turbine tankers from the Mediterranean to New York.

This size does not seem to be out of line when it is realized that before World War II, Harland & Wolff, the famous British shipbuilders, laid down a 1,000-ft. passenger liner with 200,000 s.hp. in Diesel-electric machinery. This consisted of 47 six-cylinder supercharged four-cycle type engines coupled to generators and furnishing current to propelling motors driving four screws. It is said that the outbreak of war caused the British Government to order work stopped on the ship.

As this huge motorship was never completed, and as there is no large American Diesel-electric tanker in service, we are obliged to use for some of our comparison figures, the U.S. Navy's motorship *Sperry*, which has now been operating for 12 years. This vessel is of 11,800 hp at the propellers, or almost the power required for a 12,500 s.hp, 32,000-tons dwc. Ac tanker. Her Diesels were built by the Cleveland Diesel Engine Division of the G.M.C., and her electric machinery by Westinghouse.

As the *Sperry* is a special purpose ship, she needs a lot of auxiliary power for such things as an electric furnace, machine shop, powerful cranes, cooking and heating for a huge crew, so the total auxiliary Diesel capacity is 2,500 b.hp. She has eight main engines totaling 12,800 b.hp, which with auxiliaries aggregate 14,850 b.hp.

The weight of the machinery of the *Sperry*, excluding gratings, ladders, day tanks and ship's service auxiliaries, but with twin propellers, twin shafts, shafting bearings, cables and control panels is only 492 long tons, or about one-half that of steam power. Of this total the eight main Diesel-generators weigh 185 tons.

For a 12,500 s.hp tanker it would be necessary to add a ninth Diesel-generator unit for propulsion purposes if the same type of Diesel were used. This would weigh 23 tons but she would require less auxiliary power than the *Sperry*, namely about 550 b.hp for the excitor, and about 300-350 b.hp for the ship's services, so the overall weight would be less than the *Sperry's* machinery. If the ship was heated electrically, and cooking also was by electricity, this second unit probably would have to be of 500 b.hp. However, the excitor unit would normally operate continuously at sea at 300 b.hp. It would be kept running at this load in case the tanker had suddenly to make an emergency crash stop. These main and auxiliary units would total 13,650

b.hp, and with propeller and shafting would weigh around 480 long tons.

A Diesel-electric tanker also would need a combination exhaust-oil fired donkey-boiler for heating cargo when nearing port, but power for the cargo pumps in port would be taken off one or two of the main Diesel-generating units.

While I personally lean toward the use of as many as ten to twelve light-weight main Diesel-generator units turning at 1,200 to 1,500 rpm, so that they could easily be lifted out for repair and overhaul purposes and replaced with spare units at sea, fewer and more powerful Diesels of about 850 rpm are more likely to receive consideration from oil company engineers, although one or two of the main units "out of action" would only slightly affect the tanker's speed.

As a matter of fact, multi-power units do not appear to be favored at all by oil company marine departments, and I presume that they would prefer even fewer engines than those in the *Sperry,* and at the same time take advantage of the progress in Diesel design made by her engine builders during the 12 years since she was commissioned.

Then again, the Diesels of the type powering the *Sperry* have been improved, having had super chargers added, and now give much more power with but an increase of 10% in weight. The revolution speed is 850 rpm, and these engines are capable of 10% power overload for emergencies. Therefore, six main units would be ample. Weight saving as compared with the *Sperry's* eight engines will be considerable.

A tankship owner has a variety of engines to select from, but the machinery weights would differ. Seeing that with a tanker a low-weight power plant is highly important, I would say that the design having the least weight consistent with reliable operation will enable the greatest pay-cargo to be carried effectively.

Several years ago one American Diesel builder produced a plan for a combined geared-Diesel and Diesel-electric tanker propulsion plant of 20,000 s.hp, to run on boiler and Diesel fuels, respectively. (See The Oil Forum, May 1951, pages 157-160.)

For the purpose of this discussion I will assume that the propeller speed of a 32,000-ton tanker will be about 100 rpm, because of available propulsion motors. Doubtless, it would be possible to increase the average light-and-loaded-ship-condition speed by having a propeller turning at 60 or 70 rpm, and probably without reduction gearing.

I have not forgotten that after World War I, the old Shipping Board installed Diesel-electric drive in three 11,200-ton dwc, 4,000 s.hp dry cargo ships. The propeller speed of these vessels was 60 rpm. They were the *Defiance, Triumph* and *Challenger.* On trial runs these former 10½-knot steamers touched 16¼ knots. Their loaded speed was almost 14 knots. No other ships of their tonnage-and-power have equalled their performance. Unfortunately, their Diesel engine revolutions were far too low and the weight too high. Crankshaft and other mechanical troubles developed, causing long stays in foreign ports that discouraged shipowners. However, that was thirty years ago!

Mercedes-Benz in Germany have recently manufactured about 100 marine Diesel engines of 1,300 b.hp at 1,500 rpm with a high degree of supercharging, and with a weight of less than three long tons each. The fuel consumption is only 0.37 lb. per b.hp hour. They also are developing a commercial engine that will develop 1,870 b.hp at 1,400 rpm or 2,000 b.hp maximum, on a weight of 4½ long tons, and with a fuel consumption of 0.37 lb., as well as a naval-type Diesel engine of 2,500

maximum b.hp and 2,000 b.hp continuous output at 1,510 rpm.

If any tanker owner was keen enough to equip a tankship of 32,000 tons dwc. building abroad with eight of these 1,870 b.hp engines coupled to generators (total 14,960 b.hp. or 14,017 s.hp at the propeller) he would have a main power plant of more than ample capacity, and could be assured of excellent reliability if run at 12,500 s.hp in average service conditions. Furthermore, he would have an emergency overload output of another 2,000 b.hp, making about 16,000 b.hp maximum, which could be most useful in wartime. Of course, the propulsion motor and propeller would have to have a similar excess capacity. There also would have to be about 800 b.hp in auxiliary Diesel-generator units for the excitation and ship's services. Thus the total power normally developed at full speed at sea would be 13,300 b.hp. I doubt if such machinery would cost more than geared turbines.

The daily fuel consumption, ex an exhaust-heated and oil-fired donkey boiler for cargo tank heating, would be approximately 50 tons per 24-hr day, when averaging 12,000 s.hp. The consumption of an average geared-turbine ship of 12,500 s.hp, but averaging about 12,000 s.hp is 460 bbl, or slightly over 70 tons, so the actual consumption is about 0.66 lb. per s.hp hour.

The weight of the Diesel-electric machinery, including generators, control panels, cables, and propulsion motor, plus auxiliaries incidental to the main engines, would not exceed 120 long tons, and perhaps less, compared with 750 tons for a geared-turbine steam plant of 12,5000 s.hp, this latter weight depending upon the steam pressure and temperature.

As stated, the steamship would burn 70 long tons of boiler oil per 24-hour day when at 16 knots; whereas, the Diesel-electric job would only consume 50 tons, not including fuel burned by a combined exhaust-heated and oil-fired donkey boiler heating cargo for several days on the loaded trip.

Consequently, on a round trip from the Persian Gulf to the U.S. East Coast at 16-knots in moderately fair weather —a distance of 16,800 nautical miles— the steamer would burn 3,162 tons of boiler fuel, compared with 2,187 tons of Diesel oil by the motorship. Fuel for cargo heating has not been included in either case. Time of round voyage ex two Canal transits, would be 43¾ days, Bunkers taken aboard at a Persian Gulf port.

The foregoing gives a weight saving in fuel of 975 tons, and a saving in machinery weight of 630 tons in favor of the Diesel-electric drive. Then there would be a saving of about 50 tons of fresh water, assuming that the steam tanker has efficient salt water evaporators, otherwise she would need 300 tons of water. Consequently, the motorship could carry 1,625 tons more pay-cargo per trip than the steamer. In other words, a shipowner could afford to pay a higher price for a Diesel-electric ship because for all intents and purposes he has a larger ship, although the overall dimensions would be the same. Alternatively, he could build a smaller hull at lower cost, and still carry the same quantity of pay-cargo as a larger steam-turbine ship.

With freight rates at USMC or $12.70 per ton, the higher earning power per round voyage loaded in one direction, would be $20,637 from which would have to be deducted the higher cost of Diesel oil amounting to $6,937, but on which the oil company makes a nice profit. This leaves a favorable balance of $13,760 at freight rates prevailing at the time of writing—(end of February 1955).

It is quite possible that before the end of this year freight rates might rise to USMC plus 25%, because world consumption of petroleum is expected to increase considerably. In this event the

motorship would show an advantage of $17,201 per trip; or $137,608 for eight loaded and seven unloaded one-way voyages a year. For this comparison I have taken fuel oil at Ras Tanura, Saudi Arabia, at $14.66 per ton, and Diesel oil at $27.26 per ton.

It will be realized what a tremendous economic advantage Diesel-electric power will have if and when high-speed Diesel engines burn boiler oil. Let's hope that the petroleum industry will join Diesel manufacturers in finding additives which will accelerate ignition and prevent gum and tar formation, so that the problem may be solved.

Where shall a tanker take on fuel? She could bunker at an American Port for the round Persian Gulf voyage or part bunker at both ends, so that she could load a maximum pay-cargo. Or she could partly bunker en route. But, to me, it seems foolish to re-transport for a distance of 8,300 miles from New York to Ras Tanura some 1,625 tons of fuel costing $15.20 and $22.97 per ton, respectively,—fuel which already has made one long tanker trip when it was imported from the Caribbean or Middle East—in order to bring back about 26,000 tons of Persian Gulf crude oil on which the freight rate is approximately $12.50 per ton. Bunkering for the round voyage at Saudi Arabian, Kuwatian or Bahrain ports would seem to be the most logical arrangement, especially as bunkers cost less in the Persian Gulf than in the U.S.A.

I might mention that at many ports abroad, bunkers are sold on a ton basis. There are 6.45 bbl. in each ton of heavy Bunker-C fuel, as against 7.4 bbl. per ton of Diesel oil. The tanker departments of oil companies usually are charged current fuel rates, as they operate almost like a separate company. Most oil companies make a greater profit per barrel on Diesel oil than they do on Bunker-C fuel, especially as the market price is nearly double. Therefore, it should be in oil company interests if directors and coordinators encouraged independent tanker owners as well as their own marine departments, to have their vessels equipped with Diesel-electric propulsion, especially such tankers as they charter from independents. I doubt if marine department officials take into consideration the matter of higher profits on sales of Diesel oil when figuring operating costs, as presumably their main concern is the success of their own department—not that of the bunkering or marketing divisions.

One cannot prepare a paper of this nature without discussing the important question of engine-room crews, particularly as there is much misconception prevalent. Good and conscientious Diesel operators can be obtained, and more can be trained. The U. S. Merchant Marine Academy at Kings Point, Long Island, New York, gives a four-year course in Diesel and steam power, and turns out a large number of excellent engineering officers every year.

Many of the Diesel engine manufacturers have schools for training steam men in Diesel operating lore, but oil company marine departments do not appear to have training classes. One does not hear the railroads complaining that they cannot obtain qualified engineers to maintain the thousands of 6,000 hp Diesel-electric locomotives now in service. With marine Diesel engines of around 2,000 hp each at 800 rpm, the usual operating time before overhaul is 5,000 hours.

A Diesel-electric tanker of 12,500 s.hp requires an engine-room crew of twelve, consisting of chief-engineer; first, second and third engineers, and a junior third; one electrician; three day mechanics; and three wipers. Only the chief, and three assistants need be experienced Diesel technicians.

It should be born in mind that today the operation of medium and high

speed Diesels is almost automatic. All of us have seen compressor sets running merrily on and off hour after hour in the roadway outside some building construction job, with no one in attendance. With marine Diesel-electric power the bridge officers on watch have the equivalent of "push-button" control over the propulsion machinery for maneuvering the ship without signaling down to the engine-room—a feature impractical with geared turbines.

Fairbanks-Morse, whose Diesels will power the 24,000 b.hp-22,230 s.hp electric-propelled icebreaker *Glacier,* has supplied some interesting information on an installation composed of the same 12-cylinder, two-cycle type, opposed-piston engines.

For a 32,000-ton tanker they propose eight main engines rated at 1,920 b.hp each, driving generators turning at 720 rpm, or a total of 15,360 b.hp. These eight Diesel-generator units would supply current to four synchronous propulsion motors, each of 3,250 b.hp at 600 rpm, or a total of 13,000 s.hp, less the small loss in the reduction gearing, as the single propeller would be driven at 100 rpm through gears.

While these engines are capable of giving considerably more power, they are conservatively rated for continuous operation. An overload of 10% could be given for two hours, or 16,896 b.hp. With this conservative rating there is a substantial reserve of power for unforeseen conditions involving any sub-standard operating labor and maintenance.

Excitation would be taken from an auxiliary Diesel-generator unit of 1,000 b.hp, which also would provide current for engine-room and general ship's purposes. The excitor would consist of a motor-generator set of 130 kw—this in duplicate! There also would be an emergency Diesel-generator of 7 kw. A single interlocked control would permit speed variations of all main engines. The overall transmission efficiency at full power will be 91.6% and 90.1% at half power. Power for the cargo pumps would be taken off one or two main engines.

Weight of the eight main Diesels, generators, auxiliary units, propulsion motors and reduction gears is approximately 327¾ long tons. An allowance of 5¼ tons for cables and control panels will bring the total machinery weight up to 333 tons, compared with around 750 tons for geared turbines, boilers, superheaters, distillation plant, etc., or a saving of 417 tons.

Fuel consumption is 0.428 lb. per s.hp hour; 0.434 lb. at three-quarters lead, and 0.483 lb. at one-quarter power. The latest geared-turbines, as we already have noted, have a consumption of 0.55 lb. per s.hp hour more or less, according to the steam pressure and temperature, and higher than this when the power is reduced at sea, or after some years of service. Usually it is over 0.6 lb.

TRANSMISSION AND CONSUMPTION EFFICIENCIES OF FAIRBANKS-MORSE D-E PLANT

Load	Overall Transmission Efficiency	Propulsion Engine Fuel Rate		Overall Fuel Rate*	
		Lb. per BHP-Hr	Lb. per SHP-Hr	Lb. per BHP-Hr	Lb. per SHP-Hr
Full	91.6%	0.375	0.410	0.375	0.428
¾	91.5%	0.375	0.410	0.375	0.434
½	90.1%	0.400	0.444	.400	0.483

*Includes all propulsion engines and one auxiliary generator operating at 75% load.

We will assume that the sustained power at the propeller of the F.M.-powered Diesel-electric tanker on a round voyage of 16,800 nautical miles

of 16 knots average speed for 43¾ days is 12,500 s.hp, and that she bunkers in the Persian Gulf for a round voyage. Actually the ship's speed would be governed by weather conditions. With one of the two 1,000 b.hp auxiliary units running at three-quarter load there would be a total engine output of 14,125 b.hp at the generators. Therefore, the daily fuel consumption, including auxiliary power would be slightly over 54 tons compared with 70 tons for geared turbines of 12,500 s.hp.

Consequently, the motorship would burn 2,322 tons of Diesel fuel costing $27.26 per ton (7.40 bbl.), whereas the steamship would burn 3,062½ tons of boiler oil at $14.66 per ton (6.45 bbl.). I assume that the steamship also bunkers in the Persian Gulf for the round trip.

This gives fuel bills of $63,298 and $44,875, respectively, or a difference of $18,423 in favor of steam power. However, if and when boiler oils can be burned successfully by high-speed Diesel engines, the motortanker's fuel bill on one Middle East-U.S. East Coast round trip will only be $34,040, or $10,856 less than the steam tanker, which is a mighty good reason why some of the petroleum industry's vast research resources should be devoted to the study and development of burning heavy fuels on high-speed Diesels.

Nevertheless, as the weight of the Diesel oil burned by the motor-tanker being only 2,322 tons, compared with 3,062½ tons for the boiler fuel burned by the geared-turbine vessel, there is a weight saving of 740½ tons. To this must be added the saving in machinery of 417 tons, plus 50 tons of fresh water required by steam boilers. These three items give a total weight saving of 1,157½ tons, which means that much additional pay cargo can be carried per voyage by the Diesel-electric tanker. At the current freight rate (February 1955) of $12.70 per ton, the added cargo earning power is $14,700 per round voyage, carrying cargo one way. However, from this cargo-capacity gain there has to be deducted the higher fuel bill of $18,423, on which the oil company makes a comfortable profit.

The freight rate from the Persian Gulf is not likely to remain at so very low a figure very long, because the free world's petroleum consumption is growing by leaps and bounds; also many obsolete tankers will have to be scrapped. If and when it rises to 25% above Maritime Commission rate, or $15.87 per ton, and if fuel costs remain at the current price-ratio, the single round voyage cargo-earning capacity of the Diesel-electric vessel will be $18,369 greater than that of the geared turbine. After deducting the higher fuel cost the gain per voyage would be $967.

It will be noted that in these several comparisons I have not included the fuel consumption for generating steam for heating cargo tankers. Presumably, this will be the same for both types of tankers.

In concluding this paper I am enabled to give advance information on a new Diesel-AC-Electric marine installation for which an experienced American manufacturer will shortly have the engines in production in collaboration with a well-known electrical equipment company.

An installation for a 32,000 tons dwc., 12,500 s.hp tanker consists of six main Diesel generator units of 2,215 b.hp and 1,600 kw. each, and two 1,070 b.hp, 750 kw. auxiliary Diesel-generator units, one of which is for stand-by purpose. Current from the six main units is supplied to a 2,400-volt propulsion motor of 12,500 s.hp at 101 rpm, with 10% overload at 104 rpm. for intermittent periods not exceeding two hours.

The foregoing gives 13,290 b.hp at the main generators, plus 2,140 b.hp at the auxiliary generators, or an aggregate

of 15,570 b.hp. Because only one auxiliary unit will be in operation under normal conditions at sea, there is an effective full power load of 14,430 b.hp., on which the daily fuel consumption can be estimated. I figure this to be 61 tons per day if we take a b.hp hour consumption of 0.4 lb. The engine builders however, give the figure of 700 gallons an hour, which works out at 54 tons per 24-hour day. In the calculations which follow I have used the builder's estimate.

Weights, including all equipment necessary for operating the main and auxiliary units are as follows:

MAIN DIESEL—GENERATORS, excitors, and all equipment such as air compressor, air tanks, silencer, blowers, excitor, water pumps, lubeoil pumps, starting motors, superchargers, filters and intercoolers, etc., but not ship's auxiliaries, ladders and gratings	288 long tons
Auxiliary Diesel-generators with their accessories	40 long tons
Propulsion motor	76 long tons
Total	404 long tons
Price of complete installation today	$1,612,500

The foregoing will propel a 32,000 ton dwc. tanker at 16 knots, dependent upon the lines of the hull, and efficiency of the design of the propeller.

On a Persian-Gulf-U.S. East Coast round voyage, and fully bunkering at Ras Tanura a tankship equipped with this machinery would burn 2,362 tons of Diesel oil on this 17,800-mile, 43¾-day trip. In comparison, a geared-turbine tanker of the same power and dimensions would consume 3,162 tons of Bunker-C fuel. The cost of the respective bunkers would be $54,388 and $46,355, or $8,033 in favor of the steamship.

However, the weight of the Diesel fuel is only 2,362 long tons against 3,162 tons for the Bunker-C fuel, so we get a saving of 800 tons in favor of the Diesel-electric vessel. This 800 tons represents additional pay-cargo capacity, to which must be added a saving in machinery weight of 346 tons, plus 50 tons of boiler water not needed by the motorship, or a total saving of 1,198 tons. The only engine-room fresh water needed for Diesel-electric drive is for the enclosed cooling system, and for the donkey-boiler for cargo steam heating.

With cargo freight at $12.70 per ton this 1,198 tons represent an extra earning power of $15,213 per voyage, or $121,704 if eight loaded and seven light one way voyages are made in a year. It is necessary, however, to deduct $8,033 representing the higher round voyage fuel bill. When freight rates go higher, so will the value of this extra pay-cargo go up, too.

It had been my intention to give details of an installation consisting of five light-weight American-built Diesel-generator marine units of 2,680 b.hp at 850 rpm., but the essential data have not been supplied by the makers.

TECHNICAL PROGRESS IN MARINE ENGINEERING DURING 1954

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STEAM RECIPROCATING MACHINERY

The past year, which has seen the disappearance of the largest steam reciprocating engines, has witnessed the successful trials, and introduction to service, of a most interesting combination of reciprocator and turbine. The engines of the *Maloja,* while not of the same size as the old giants in the *Olympic* or *Kaiser Wilhelm II.* (40,000 I.H.P.), were of quite notable dimensions, having 97-in. diameter 1.-p. pistons and a crankshaft of 18½ in. diameter, and for many years they have been the biggest reciprocators under the British flag. The passing of the *Maloja* to the ship-breakers closes a chapter in the history of steam engines.

In contrast, the machinery of the *Baron Ardrossan* is an example of the best present-day practice in steam reciprocating installations, which has resulted in a fuel-oil consumption, for all purposes, of 0.745 lb. per S.H.P. per hour on loaded trials, when developing 2,378 H.P. at 78 r.p.m. This remarkable figure is attained by carefully matching-up an N.E.M. Reheat triple-expansion engine with a Bauer-Wach exhaust turbine. The equivalent I.H.P. for normal service is 2,650 at 75 r.p.m., with an overload capacity up to an equivalent of 3,050 I.H.P. In both cases, the exhaust turbine delivers a little less than one-quarter of the total output. Steam is supplied by two N.E.M. Scotch boilers at a pressure of 200 lb. per sq. in. and a temperature of 750 deg. F. A condenser vacuum of 28½ in. mercury was obtained on trial.

Particularly noteworthy is the ingenuity shown in devising an accurate means of measuring the rate of condensate discharge during the loaded trials. For this purpose, one of the main feed pumps was used to pump from the hotwell into a storage tank of 3,000 lb. capacity, which discharged into a measuring tank, and which, in turn, discharged into a second storage tank of the same capacity. The second feed pump drew from this tank and discharged to the boiler. The measuring tank had quick-acting valves, 6 in. in diameter at each end, was fitted with a gauge glass, and had been carefully calibrated. In this way, suitable operation of the valves enabled the whole of the condensate to be measured continuously and accurately. During trials, the drain discharges and the fuel oil were also continually metered.

The results of these trials will do much to revive interest in the reciprocating steam engine for moderate powers, and the results of service experience are looked forward to with interest. A daily consumption of 19 tons of fuel for a 2,350-S.H.P. installation, with all the advantages of steam-engine robustness and reliability, is an achievement of which to be proud.

Retarders fitted to fire-tubes are known to be effective in improving the rate of heat transmission from gas to water, but, when fitted, they are often discarded at the first boiler cleaning, because of the difficulty of cleaning and the trouble of refitting. Tests, recently carried out by an independent authority, have shown that a plain tube, fitted with retarders, has a better performance than any of the peculiarly shaped tubes, of equivalent hydraulic diameter, which are available. This fact, together with the development of an air lance, using percussive blasts of air capable of cleaning tubes with retarders in place (it will even descale them), may revive the practice of fitting this much-maligned, but most effective, means of reducing the losses due to high temperature of uptake gas and of improving boiler efficiency.

STEAM TURBINES

Tanker power plant still continues to increase in size, as, for example, the *Tina Onassis* (45,000 tons deadweight), the machinery for which develops 17,000 S.H.P. and gives a speed of 17 knots, and the *W. Alton Jones* (38,000 tons deadweight), with machinery developing 22,000 S.H.P. for a speed of 17½ knots. Both of these vessels are driven by a single screw. Fully tabulated results of the *Tina Onassis* trials have been published, together with several graphs and notes concerning her maiden voyage. The fuel consumption, on a 7-hour run at 16,600 S.H.P., was 0.531 lb. per S.H.P. per hour, the boiler efficiency being recorded as 93.2 per cent on the lower calorific value of the fuel, equivalent to about 86.5 per cent on the gross calorific value. The fuel-consumption figure can be considered very good, in view of the steam conditions which, at 625 lb. per sq. in. and 833 deg. F., are not unusually high. An equally good fuel consumption is claimed for the *Mare Adriacum*, of 31,600 tons deadweight, and 16,000 S.H.P. at 18 knots.

The gap between 0.53 lb. per S.H.P. per hour and the best fuel-consumption figure for land practice, which is 0.43 lb. per S.H.P. per hour, is mainly due to the larger size of land power-station installations, with consequent reduction of parasitic losses, their more advanced working conditions, the greater use of reheating and of regenerative feed-heating, and the arrangements made for the generation of auxiliary power. In shore installations, the "house" generator is often driven by a main turbine, operating in tandem with a main generator. A consideration of these factors clearly marks the path towards improved efficiency of marine installations.

This aspect is particularly dealt with by Mr. T. W. F. Brown, D.Sc., S.M., in the Twenty-sixth Thomas Lowe Gray Lecture to the Institution of Mechanical Engineers, on the subject of "High-temperature Turbine Machinery for Marine Propulsion." In this, the lecturer examines present practice in steam-turbine machinery for both ship propulsion and land power generation. An appendix lists materials and their properties recommended for various items, and which effectively determine the temperature limit that can be used in practice.

Within the practical temperature range, taken as 1,500 deg. F., improvements due to pressure advancement and reheating are assessed. It is pointed out that reheating can be effected by the use of live steam, boiler gas or some exchange medium, such as liquid metal, as will be used in the atomic generation of steam; this latter method is favored.

In discussing problems in connection with the development of high-efficiency gas turbines, it is pointed out that the high-temperature properties of materials are again a principal limiting factor. For open-cycle gas turbines, this factor controls the material used for h.-p. rotor blading, and in steam turbines it controls the superheater material. Whereas various methods are available for cooling gas-turbine blades, there are no such means available for cooling superheater tubes, and, in fact, as they form part of the necessary path of heat-exchange between fuel and working fluid, any artificial means of cooling is impossible. The same limitation is imposed on the closed-cycle gas turbine by the tube material of the high-temperature air-heater. Thus, a natural limit is set to the maximum temperature of the working fluid in the steam-turbine cycle and the closed-cycle gas turbine, which does not apply to the open-cycle gas turbine.

Two extremely interesting papers, which record operating experience with high-pressure, high-temperature turbine installations, appeared towards the end of the past year. An account of two years' experience of the operation of the *Nestor* and *Neleus*, the first two British ships to operate at a temperature of 950 deg. F., is given by Mr. L. Baker, D.S.C., and Mr. W. H. Falconer in their paper entitled "Some Operating Experience at 950 deg. F.," read before the Institute of Marine Engineers in November last.

A group of five papers was presented at the annual meeting of the American Society of Naval Architects and Marine Engineers as a "Symposium on Turbine-operating Experience." Part II. of the symposium describes certain unsatisfactory operating experiences which occurred during the trials or early life of turbine sets, designed to operate at a temperature of 1,000 deg. F.

The conclusions reached in both this and the British paper are particularly encouraging. Certain of the conclusions in connection with the *Nestor* and *Neleus* are quoted—"(1) No difficulties have been experienced due to operation at 950 deg. F.; (2) no difficulties have been experienced due to rapid temperature fluctuation from 950 deg. F. to 750 deg. F., or *vice versa;* (3) no difficulties have been experienced due to the greatly reduced turbine clearances; and (4) no important defects have arisen with the dampers of the boilers."

An investigation into the effect of tip clearance on the efficiency of turbine blading is reported by Mr. J. L. Jefferson, B.A., in a paper, entitled "Some Practical Aspects of Tip Clearance in Turbine Blading." In this paper, which was read before the North-East Coast Institution of Engineers and Shipbuilders, static tests are described and their limitations are commented on, but most of the results quoted have been obtained under running conditions, using an experimental turbine adapted for both single and multi-stage blade arrangements. It is concluded that the use of shrouding in multi-stage turbines should allow a greater latitude in the choice of pitch, due to its effect in reducing the rate of efficiency loss incurred by the use of excessive pitch, although some of this advantage is lost if blades are worked at low-velocity ratio. With radial-clearance blading, it is shown that profiles having a large pitch-chord ratio can be very sensitive to clearance.

For the case of single-row wheels, the effects of degree of reaction and of disc ventilation are illustrated, and the effect of clearance over a wide range of incidence is shown for different blade profiles.

BOILERS AND STEAM PIPES

It is reported that the Velox boilers installed in the *Ville-de Tunis* and the *Cambodge* are continuing to operate successfully, and were found to be in a satisfactory condition when opened up several months ago, by which time the *Ville-de-Tunis* had operated for about 5,600 hours, and the *Cambodge*, 2,100 hours. Deposits have been detected on the heat-transfer surfaces, and a regular routine is now instituted for washing-out the boiler and cleaning the gas passages every 500 hours. Three men take six hours for this work.

The boilers for six new 18,000-ton oil-tankers for the Eagle Oil & Shipping Co., Ltd., of London, are to be of La Mont forced-circulated design. The normal full-power evaporation of each boiler will be 50,000 lb. per hour, from an economizer inlet temperature of 320 deg. F. to super-heater stop-valve conditions of 500 lb. per sq. in at 800 deg. F. There are 2,039 sq. ft. of heat-transfer surface for generation, 1,250 sq. ft. for the superheater, 1,668 sq. ft. for the economizer, and 6,715 sq. ft. for the air heater. The use of forced draft only, somewhat simplifies the control and layout of the boiler, and, as a further simplification, it is intended to maintain a constant fuel-oil pressure, even while maneuvering. This is likely to ensure uniformly good combustion over part-load working, and the desired flexibility is introduced by using five oil burners in each boiler, three of which have 14-in. diameter registers, one a 12-in. register and the remaining one a 9-in. register, the necessary adjustments being made by selection of burners rather than by variation of oil pressure. In the *San Florentino*, consistently good combustion has been obtained by this means, with a fuel-oil pressure in excess of 200 lb. per sq. in.

A glandless boiler-circulation pump, driven by a submerged 15-H.P. motor, is used with each boiler. The circulation rate of the pump is 134 tons per hour, which gives a circulation ratio of 6:1 within the boiler. The overall efficiency of the boiler is reported as 87.18 per cent.

A formula for the calculation of pressure drop, in the tubes of forced-circulated steam boilers, is developed by Mr. J. C. Koel in an article appearing in *Ingenieur* (The Hague) of the 26th March last. Cases of uniform and of non-uniform heat-absorption rates are considered, and, in particular, the stability of flow in the parallel tubes of a La Mont boiler is considered. Briefly, stability is found to be independent of the circulation ratio; it is reduced considerably by cold feed-water, and improved by the use of hot feed; an uneven heat load reacts unfavorably on stability of flow; but this disadvantage can be countered by fitting appropriate inlet orifices to each tube. An equation is given for calculating the resistance of such orifices.

In the endeavor for high overall efficiency of boiler-turbine installations, the maximum use of regenerative feed-heating greatly aids turbine efficiency, but necessarily reduces the heat-recovery capacity of the economizer; so that on a high-duty boiler, the air-heater surface becomes comparatively large, in order to raise the component efficiency of the boiler. In addition to its bulk, a further drawback is the possibility of the flue-gas temperature at the cold end being depressed below the corrosion danger

limit, especially when sulphurous residuals are burnt, with consequent troubles due to leakage and by-passing.

The recent and successful application of bled steam for air-heating would seem to offer a suitable means of overcoming some of the air-heater drawbacks. There will be many installations in which it could be used as the first stage of air preheating, in series with a flue-gas preheater, in which case, the size of the latter could be reduced, while, at the same time, the exit flue-gas temperature would be raised. Raising this figure much above 350 deg. F. (taken as corrosion limit) results in a loss of boiler efficiency at the rate of about 1 per cent for each 40 degrees.

Marine turbines are seldom more than medium sized, and the use of bled steam for initial air heating will always be beneficial in increasing the steam-flow rate through the h.-p. blading, as well as by the intrinsic gain due to the recycling of a certain proportion of the heat released by the fuel. The use of two-stage air preheating (bled-steam heating followed by flue-gas heating) would thus seem to merit consideration, especially where bled-steam heating of combustion air alone, results in an uptake temperature which is too high.

A portable boiler usually conjures up a picture of a locomotive boiler on wheels, but, at the Royal Canadian Naval Dockyard, Halifax, Nova Scotia, there is being used a different type, although it is still an adaptation of a kind of locomotive boiler. Passenger-traffic Diesel locomotives are fitted with a very compact, automatically-controlled, steam boiler for supplying accommodation heating in the train. The R.C.N. Dockyard is using such "packaged" steam generators for a variety of purposes where steam is required at a pressure of 70 to 300 lb. per sq. in. at a rate of up to 5,000 lb. per hour. Steam can be raised from "boiler cold" in two minutes, during which period compressed air is required for fuel atomization. After this start-up period, steam is used for atomization, and the boiler becomes completely and automatically controlled.

The units are portable, and are moved about the dock on fork-lift trucks. They are employed for operating steam cranes and pile drivers, and for testing sea-water evaporators and other equipment.

When considering the values of allowable stress and weld efficiency to be used in the design of welded pressure vessels, both fired and unfired, each code seems to have its own particular value, which, in the case of stress, ranges from 27 per cent to 66 per cent of the yield stress. This position is examined and discussed by Mr. W. A. Kerkof, in an article entitled "Stresses for Pressure Vessels and Boilers up to 650 deg. F.," which appeared in *The Welding Journal* for May, 1954. The discussion is limited to vessels required for temperatures below 650 deg. F. so that creep phenomena can be excluded, and the analysis is further simplified by the following assumptions: — That the material follows Hooke's law up to the yield point, that the yield point of the weld material is that of the parent metal, that there is no strain hardening, and that, under conditions of combined stress, yielding occurs when the maximum stress attains the value of the yield-stress in simple tension.

Residual stresses are discussed, and it is concluded that the stress-relieving of welded pressure vessels by heat-treatment is unnecessary; as unnecessary, in fact, as it is for riveted boilers. It is pointed out that, since the high total stresses that occur in non-stress-relieved vessels are permitted, it is considered that the expense of heat-treatment is unnecessary to relieve such stresses, but it may be required

to create improved mechanical properties, such as, for example, hardness and notch sensitivity in low-alloy steels. The hydraulic test has a marked stress-relieving effect, the magnitude of which depends on the ratio of hydraulic test pressure to working pressure.

It is reasoned that stress concentration factors should not be greater than two, and it is pointed out that the triaxial stress concentrations, which occur at such places as branch connections and manholes, will only be high as a result of faulty design. Only in exceptionally thin walled vessels is out-of-roundness found to threaten dangerous increase of stress concentration. In the case of other vessels, the permissible out-of-roundness, based on stress concentration danger, is greater than that which could be tolerated for practical reasons.

From a study of a number of widely used codes in regard to non-radiographed, non-stress-relieved pressure vessels, it is suggested that the weld efficiency be taken as 100 per cent, and that the allowable membrane stress be reduced to two-thirds that of the yield stress in simple tension. It is also suggested that the basis for calculating knuckle radius and dome height of dished ends should be based on the same allowable membrane stress of two-thirds the simple yield stress, the superimposed bending stress being limited to the same figure.

The practice of hydraulically testing vessels to the point of failure is criticized, on the ground that the vessel yields completely during such a test, and is in a completely artificial state.

When considering the scantlings of steam pipes for working temperatures in excess of 1,000 deg. F., there is no general agreement among the standard codes as to what should be the criterion of material strength for the purposes of design. In an article entitled "High-temperature Steam Pipes—Properties of Steel and Basis of Design," which appeared in *Engineering* (9th April, 1954), Mr. P. H. Margen, B.Sc. (Eng.), discusses this problem, and makes a plea for an agreed framework, whereby working stresses can be computed from available test data, with due allowance for errors in extrapolation of short-time tests and for variations of quality. The properties of the principal pipe materials used in Great Britain, for service temperatures above 1,000 deg. F., are discussed, and the proposed working stresses are applied to some examples of design.

It is suggested that it is an unnecessary restriction to use creep strain or creep rate as the criterion of pipe design (as most codes do). Pipes are not precision components with fine clearances to be guarded, but are items which normally undergo considerable movement and distortion due to thermal contraction and expansion. Thus, it is suggested that, for high-temperature pipes, the allowable working stresses should be based on (*a*) the rupture strength over a reasonable expectation of steaming life, and (*b*) a "rupture safety factor" which will make allowances for variations in surface deformations and deviations from designed working conditions, such as those covered in B.S.132 for specification of turbine material. Within these limits of operational deviation, a table is given showing a suggested method of specifying the working stress which should be used in calculating wall thickness by means of the A.S.M.E. 1952 Code formula. Corrosion cannot be neglected, and the allowance varies with both temperature and type of steel. It is pointed out that corrosion becomes appreciable for steels of low chromium content at temperatures above 1,050 deg. F.

A controlling consideration, which, of course, must not be lost sight of, is the condition which occurs when high-temperature piping cools down after suffering considerable plastic deforma-

tion. The reversed high-temperature strains have to be withstood by the material with its "cold" properties of elasticity and ductility.

The case for considering the rupture strength of piping materials, as the design criterion for high-temperature conditions, has also been put forward by Mr. J. S. Blair, in an article entitled "Calculation of the Thickness of High-temperature Steam Pipes," which appeared in *Engineering* (30th October and 4th December, 1953). It is argued that, owing to the relatively rapid equalization of stress across the wall of the pipe due to creep, the maximum stress falls from that which is estimated for elastic conditions, to that given by the thin-cylinder formula, and that, in fact, these stresses can be used as being representative of the stress condition throughout the life of the pipework. Pipe material must show appreciable elongation and ductility prior to rupture under long-term test conditions, if such assumptions are to be valid. It is suggested that minimum thickness be based on a thin-cylinder formula, using an allowable stress which is 60 per cent of that which would cause rupture in 100,000 hours at the design temperature.

Mr. F. van Iterson, in discussing this article, contended that it is now widely accepted that considerations of plastic deformation should govern pipe-wall thickness, and he gives an alternative formula for use.

CONDENSERS

The failure of condenser tubes is not now the problem that it was some years ago, thanks largely to the classic work carried out by Mr. G. D. Bengough and his many collaborators for the Corrosion Committee of the Institute of Metals during the period 1910 to 1930, and which was continued under Mr. R. May when the work was transferred to the aegis of the British Non-ferrous Metals Research Association.

Compared with the position 30 years ago, the state to-day is radically changed; a premature failure is an unusual and noteworthy event, and it is normally expected that condenser tubes will last the whole life of the ship. There is no longer an overall condenser-tube problem, but there are still a small number of individual problems, which affect a very small percentage of the tubes in service. Up-to-date information and a survey of the subject are given in the paper entitled "The Resistance to Failure of Condenser and Heat-exchanger Tubes in Marine Service," read before the Institute of Marine Engineers, some little time ago, by Mr. P. T. Gilbert, Ph.D.

A brief account is given of the various condenser-tube materials at present available, and of the types of corrosion to which they are resistant. The factors responsible for the small number of failures that still occur are listed and discussed. Of the cases investigated, since 1945, by the British Non-ferrous Metals Research Association, 27 per cent concerned merchant vessels, 50 per cent naval vessels and 23 per cent power stations; while of the total cases, 60 per cent occurred in main-turbine condensers, the remaining 40 per cent being in turbo-generator condensers, drain coolers, oil coolers and other heat-exchangers. Brass tubes were involved in 85 per cent of cases (55 per cent being arsenical aluminum brasses and 30 per cent other brasses); the other 15 per cent involved 70/30 cupro-nickel tubes.

The factors shown by experience to be the most important are listed and dealt with in detail. Foremost among the reasons for failure is the use of polluted cooling waters, particularly those containing sulphuretted hydrogen, sulphur or organic compounds,

which are particularly liable to affect aluminum-brass tubes. It is particularly important not to allow polluted waters to remain in condenser tubes (especially those of aluminum brass) for any appreciable length of time, during the early life of the ship, because the necessary protective films which form when the system, as a whole, is immersed, take an appreciable time to become established.

Another potent source of trouble is the incorrect use of sacrificial protector blocks in water-boxes. In explaining the mechanism of protection, it is pointed out that, once protection is started, it must be continued effectively, as otherwise, if started and then discontinued, the tube ends are liable to corrode much more rapidly than they would have done, if protection had never been attempted. Zinc blocks are unsuitable for protection purposes as they are, initially, highly active. Subsequently, however, they become polarized and inactive long before they disappear, thus inducing a highly dangerous state which is not immediately apparent to a casual inspection. Steel blocks remain active until they completely disappear, and the presence of iron corrosion products in the water appears to assist in the formation of a good protective film on the tube surface. They must, of course, be properly maintained and replaced before there is danger of their coming loose and being displaced. A displaced block can be a source of danger.

Corrosion of cast-iron and steel water-boxes can be prevented by correct painting, properly maintained, but, to ensure the latter provision, it requires more frequent attention than is usually given. Recently, it has been demonstrated that corrosion in water-boxes can be prevented by the installation of sacrificial magnesium anodes, and there is no evidence, so far, that such are detrimental to the performance of condenser tubes, with the important proviso that they are renewed before exhaustion occurs.

In a short comment on heat-transfer rates, it is pointed out that, if aluminum brass is suitable for other reasons, it will have the additional advantage of having a considerably lower resistance to heat transfer than has cupro-nickel—a fact borne out by practical experience.

An interesting device for measuring heat flow was described in *Engineering* (July, 1954). This device, the Hatfield heat-flow meter, is manufactured by Messrs. Joyce, Loebl & Co., Ltd. Discs of tellurium-silver alloy, coated with copper, act as a thermocouple, and the dimensions (1 cm. diameter by 1.5 mm. thick) are such that a heat-flow rate of 1 B.Th.U. per sq. ft. per hour generates about 2.5 microvolts. They can be either completely embedded in the material through which the heat-flow rate is required, or they may be clamped to the surface from which heat loss occurs. They can be used for estimating heat-flow rates or heat losses on many items. In tests carried out on cold-storage chambers, the values for heat leakage, estimated by means of disc measurements, agreed within 3 per cent with those obtained from an energy balance.

DIESEL MACHINERY

A useful summary of the factors influencing the use of heavy oil in Diesel machinery was given in an article by Mr. H. T. Winberg, entitled "The Purification of Heavy Fuel Oil for Diesel Engines," which appeared in the July Number of *The Motor Ship*. The value of the price differential, the extra cost of the installation, and the ability of separator plant to purify oil are examined and discussed in the light of recent experiments and data. The general conclusions reached concerning the suitability of using heavy oils

for Diesel machinery are that the suitability decreases with increasing engine speed; four-stroke cycle engines are more suitable than two-stroke cycle engines; single-acting engines more suitable than double-acting engines; supercharged engines more suitable than those having atmospheric induction; a high uniform load more suitable than a varying load; and an efficient sealing diaphragm between cylinder and crankcase greatly increases the suitability.

The higher rates of cylinder-liner wear, which accompany the use of heavy fuels, can be reduced by the use of satisfactory heating and purification plant, high-temperature cooling water, chromium-plated cylinder liners, and detergent lubricating oil with adequate purifying capacity. This latter point, concerning the adequate purification of lubricating oil, is shown, by many unfortunate experiences, to be vital.

The performance of four vessels, all powered with Doxford engines and burning a variety of fuels, has been examined and reported by Mr. H. F. Jones, Mr. D. Royle and Mr. R. G. Sayer, in their joint paper on "Fuel Features related to Operating Experiences in Motorships using Low-cost Fuels," presented before the Institute of Marine Engineers in December, 1954. Samples of the fuels bunkered were retained from the three cargo vessels operating from the United Kingdom, two of which were engaged on a Mediterranean route, with the third running to Australia. Certain information was also available from a tanker operating mainly between the Gulf and American North-East Coast ports. Sludges from purifiers were also sampled, and the various analyses are reported, together with summaries of the conditions of piston rings and the wear rates. In certain instances, lubricating oils were varied, detergent and non-detergent types being used.

The considerable amount of information in the paper makes it difficult to summarize, but it is interesting to note that the comparison of liner-wear rates with detergent lubricating oils and non-detergent oils reveals no significant difference, although there is a slight trend in favor of the non-detergent type. There have been indications that combustion is impaired if the amount of water which remains in the fuel after centrifuging is greater than 0.5 per cent. The emulsions formed between water and residual fuels are comparatively stable, and are sometimes difficult to break-down. Conradson carbon value is not considered to be a critical factor, if it is less than 14 per cent, and, provided the injection is suitably advanced, fuels containing up to 8 per cent asphaltenes can be successfully burned. The effect of sulphur content is still controversial; but it is quite certain that, through inefficient combustion of sulphur-bearing fuels, resulting in sludge deposits on pistons, rings and scavenge ports, a train of events can be set in motion leading to heavy crankshaft corrosion. This train can be broken by preventing the contamination of the lubricating oil with sludge deposits from imperfect combustion, and, in the absence of such a safeguard, the severity of the attack can be considerably reduced by the proper use of adequate lubricating-oil cleaning devices, together with the use of alkaline additives to the piston-cooling water.

Sulphur attack is sometimes suspected on cylinder liners. For a variety of reasons, the temperature of the cylinder-jacket circulating-water discharge cannot be a reliable guide to the temperature on the gas face of the liner, and, therefore, there is no direct means of knowing whether the sulphuric-acid dew-point is reached or not. Etched spots on cast-iron liners quickly become abraided, and may thus often be disguised on opening-up

a cylinder for inspection. The sulphuric-acid marks on porous chrome-plated liners, however, do not abraid, and, on such liners, etched spots known as "milky patch" are readily recognizable. Such marks provide a reliable indication of the presence of sulphuric acid forming conditions and enables the discharge temperature of the cooling water to be adjusted to eliminate, or at least to reduce, the attack.

Several proprietary additives are now appearing for use in connection with burning heavy fuels in Diesel machinery. One such additive, manufactured by Xzit, Ltd., is said to be capable of reducing sludge content of oils, and also, due to its effect on lowering combustion temperature, the likelihood of carbon deposition within the cylinder is greatly reduced, as will be the attendant wear rates. Recommended dosage is dependent on the conditions normally encountered in a particular engine, and the cost of treatment, it is stated, is likely to vary between 1s. 10d. and 2s. 4d. per ton of fuel treated.

Similar improvement in sludge reduction, and in combustion conditions, is claimed for an additive manufactured by Gamlen, Ltd., the estimated cost being about 1s. 0d. per ton of fuel. A further advantage claimed for such additives is that the heat-transfer surfaces of waste-heat boilers remain in better condition and require less frequent cleaning.

Considerable developmental work in connection with supercharging has come to fruition at the works of Nydqvist and Holm, A/B., who now have in production their exhaust-gas turbo-charged E-type engines. For marine purposes, without increase in crankshaft size, the increase in power due to supercharging is limited to 30 per cent in order to comply with classification society regulations. The engine develops 91 S.H.P. per cylinder at 700 r.p.m., with an M.E.P. of 109.5 lb. per sq. in.

In the range of bigger machinery, additional information has become available, during the past year, concerning the design and performance of the Stork turbo-charged two-stroke cycle engine installed in the *Ouwerkerk*. The machinery develops 8,200 S.H.P. at 115 r.p.m., with a fuel consumption of 0.328 lb. per S.H.P. per hour. The bore and stroke are 750 mm. and 1,500 mm., respectively.

The eight-cylinders are arranged in four groups of two, and each pair of cylinders exhausts into a Brown-Boveri V.T.R.500-type turbo-blower. Scavenging is on the uniflow principle; the exhaust is through four valves arranged symmetrically in the cylinder head, and the inlet through piston-controlled ports in the cylinder liner. The exhaust passages between the cylinder head and turbo-blower are very carefully stream-lined and made as short as possible, so that the full effect of the kinetic energy of the exhaust blow-down pulse is available for use in the turbine, as well as the energy due to temperature and pressure of the exhaust gas. In this engine, the cranks of each pair are mutually at 180 deg.; but work at Delft University indicates that the interval between pulses from two successive cylinders can be reduced to a minimum of 120 deg., without the scavenging of one cylinder being disturbed by the exhaust-pulse pressure of the next. Thus, it would appear possible that three cylinders could be hooked on to one turbo-charger and still make good use of the blow-down energy available.

The adoption of multiple blowers means that a well-proven standard type is used, and, furthermore, it has been demonstrated that the engine will run at 85 per cent r.p.m. (and corresponding propeller loading) with one blower stopped, and still show a clear exhaust.

On starting with the turbo-blowers stationary, the blow-down pulse from

the starting air is sufficient to bring the turbine speed from zero up to 2,000 r.p.m.—a speed which is more than adequate for slow running, and from which full power can be built up just as quickly as with the non-supercharged engine. Slow running can be maintained at 18 r.p.m. with clear exhaust, under which conditions the blowers are rotating at just slightly over 1,000 r.p.m.

Sea-water coolers are arranged to cool the scavenger air between blower discharge and scavenge trunking, in order to improve the supercharge. Heat-balance records indicate that the thermal load on the piston, rings and jackets is no more severe at an M.E.P. of 92.5 lb. per sq. in. when supercharged, than at 75.5 lb. per sq. in. when unsupercharged.

On this particular engine, the pistons, valves and running gear of the normal scavenge pumps have been removed, and in future engines they will not be installed.

Forschungshefte fur Schiffstecnik for March, 1954, contained the completion of a statistical analysis by Mr. A. Hansen, based on data obtained from about 1,400 motor vessels, of oil-tank and other types, with deadweights ranging from 2,000 tons to 32,000 tons, all of which were built in 1952. The data are used to derive empirical relationships, shown in graphical form, between the following factors—(*a*) Cylinder bore and stroke, piston speed and r.p.m., number of cylinders, type of drive and number of engines, or (*b*) propeller efficiency, speed, dimensions, weight and cost of installation, running costs and reliability.

A general conclusion is arrived at that there are economies of space, weight and cost of installation to be obtained from the use of multi-engine installations with indirect drive, which become more significant as the size of vessel increases. Under certain conditions, there is also a reduction of time lost in maintenance for such installations.

In view of the continued interest in geared-Diesel drive, an early example is brought to mind in the *Fairsea*, built in 1941 by the Sun Shipbuilding & Dry Dock Company, of Chester, Pa., and propelled by twin Sun-Doxford engines. Each engine is rated at 4,500 S.H.P. at 180 r.p.m. with a permissible overload of 25 per cent., *i.e.*, 5,625 S.H.P. at 195 r.p.m. The single screw, normally running at 75 r.p.m., is driven through Westinghouse electric slip couplings and a single-reduction gear of 2.14:1. The vessel was used by the United States Navy as an aircraft-carrier during the war, and, since 1949, it has been used as an emigrant ship between Bremerhaven and Australia, in which service the vessel has averaged 330 days at sea per annum (the mean service speed being 16 to 17 knots) without engine trouble of any consequence, either in the main engine or transmission. A distillate fuel is used.

A hydraulic device has been developed under the name of the De Schelde synchronizer, which can be used to control the speed and relative position of the crankshaft of two oil engines. It permits the phase relation between the two shafts to be varied between zero and 360 deg. while they are running, and any desired value of phase angle will be maintained constant.

The system is comprised of two main parts, the hydraulic differential, which reacts to change of relative position of its components, and the hydraulic control of the fuel-pump systems, which reacts to signals from the hydraulic differential. The principal components of the differential are two cylindrical slides rotating one within the other, which are both driven in the same direction, one by each engine. Any difference in the r.p.m. of the two engines results in the relative movement of the two slides. With both engines at the same speed, and with the desired phase

relation, ports in the two slides are closed; but as one gains on the other, one set of ports is uncovered, and servo-oil is admitted to a mechanism acting on the fuel pumps of the faster engine, in such a way as to slow it down. There is thus no danger of overloading an engine. When once more synchronized, the servo-oil is cut-off from the fuel-pump control, and the r.p.m. returns to normal. The twin-screw vessel *Kungsholm* is fitted with this device to control the engines, each of which develops 7,000 S.H.P. at 115 r.p.m.

The versatility of the Diesel engine is such that there are several installations in which Diesel machinery even drives paddle-wheels. The Admirality have recently placed a contract for two Diesel-electric paddle-wheel tugs. Such a method of driving can be quite flexible, and may be either through two electric motors, which are coupled to their respective paddle-wheels by belting, chain or gears, or one electric motor may be used with either of these methods of drive. Very heavy control gear is required if the motor is coupled directly to the paddle-shaft.

The use of nodular cast-iron for crankshafts is discussed in detail by Mr. S. B. Bailey, M.Sc. (Eng.), in a paper entitled "Nodular Cast-iron—Its Present Position and Future Prospects as an Engineering Material," read before the Institution of Mechanical Engineers in April, 1954. The same material is mentioned by Dr. L. B. Pfeil, O.B.E., F.R.S., in his paper entitled "New Metals in Engineering," presented before the Institute of Marine Engineers in the same month. The toughness, ductility and impact resistance, which can be developed in nodular irons by simple heat-treatment, make them suitable for many parts where shock resistance is a primary consideration. The principal advantage over malleable irons is the shorter length of heat-treatment that is required. As a replacement for carbon steels, nodular cast-iron is attractive for components which are difficult to cast, which require high surface hardness with medium strength and impact resistance, or where low-notch sensitivity is especially required.

The danger from torsional vibrations can be reduced either by detuning or by increasing the damping capacity of the system. Detuning is effected by the introduction of non-linear stiffness; typical examples are spring or bonded-rubber couplings providing a variable natural frequency, which can prevent the establishment of dangerous states of resonance. The introduction of increased damping capacity has the effect of absorbing some of the input energy and converting it to heat (by way of eddies or friction), and thereby reducing the amplitude of vibration at resonance. The viscous fluids available until recently were not completely satisfactory for such work, on account of the reduction of their viscosity with rise of temperature, their instability and their tendency to break-down under continued mechanical shearing. Synthetic silicone fluids, however, do not suffer from these disadvantages. They are stable, non-corrosive and have a flat viscosity-temperature characteristic, and are eminently suitable for filling viscous dampers.

The Admiralty Experimental Laboratory have designed such a damper, consisting of a casing, rigidly attached to the free end of the crankshaft to be protected, and a large inertia mass, located on a phosphor-bronze bush carried on the casing. The small clearance spaces between the casing and the mass are maintained full of a synthetic silicone fluid; so that, at critical speeds, the heavy mass tends to rotate at a uniform speed, and the casing, vibrating with the shaft, will be oscillating relatively to it, and will be continually

shearing the separating silicone film. The energy absorbed protects the shafting and is dissipated in the form of heat.

GAS TURBINES

It seems to be generally accepted that, for the present time, the long-life marine gas turbine is limited to an inlet temperature of about 1,250 deg. F., a temperature which can be tolerated with only a relatively small amount of disc and bearing cooling, the blades being permitted to run at gas temperature. This temperature limitation also reduces the danger due to vanadium-pentoxide attack when burning residual fuels, and, in view of the comparative success of the Diesel engine in using such fuels, it is essential for the long-life gas turbine to do the same. It is of interest, at this point, to compare figures for fuel consumption of various types of main engines, and the figures given in the accompanying Table indicate the performances with which the gas turbine has to compete in the various power ranges.

Steam reciprocating engine, reheated with Bauer-Wach exhaust turbine.....	2,650 I.H.P.	0.745 lb per S.H.P. per hour.
Steam turbine; 425 lb per sq. in. at 750 deg. F.....	8,000/10,000 S.H.P.	0.72 lb per S.H.P. per hour.
Steam turbine; 900 lb per sq. in. at 900 deg. F.....	15,000 S.H.P.	0.53 lb per S.H.P. per hour.
Diesel engine; normally aspirated.....	8,000 S.H.P.	0.37 lb per S.H.P. per hour.

Of existing long-life sets, those using free-piston gas generators have consumptions which most nearly approach the low Diesel figure. As was recorded in 1954, there is, in France, considerable interest in propulsion by means of exhaust-gas turbines fed from such gas generators. Twenty-one minesweepers and two coasting ships were projected, and the first vessels have been placed in service.

The coasting ship *Cantenac* has been built by Chantiers et Ateliers Augustin-Normand, of Havre, and engined with Sigma-Seine gas generators. These generators supply Alsthom gas turbines, which drive through reduction gears, reducing from 9,000 r.p.m. to 220 r.p.m. The gas generators are capable of burning residuals up to 900 secs. Redwood No. 1, at 100 deg. F., after suitable heating and cleaning, although they are started-up on distillate fuel. A recent test carried out on somewhat similar machinery gave a specific fuel consumption of 0.57 lb. per S.H.P. per hour at one-third full load, and 0.44 lb. per S.H.P. per hour at full load, which, in this case, was 1,050 S.H.P.

A second vessel, the *Merignac,* is equipped with Pescara-type free-piston gas generators, exhausting to the turbines at 43 lb. per sq. in. (gauge) and 825 deg. F. A single-wheel astern turbine is integral with each ahead turbine, and by means of suitable screening, the windage loss has been reduced from its originally estimated figure of 100 H.P. Maneuvering has proved to be reasonable, taking 50 seconds to stop the propeller from 220 r.p.m. ahead, and a further 34 seconds to work up to 100 r.p.m. astern.

The installations are noisy, but, in view of the rapid development of this

type of machinery, there is little doubt that such outstanding problems will soon be solved.

A paper entitled "The Free-piston Gas-generator as a Power Plant for Ship Propulsion," was read by Mr. F. A. I. Muntz, B.A., and Mr. R. Huber, Dipl. Ing., before the Institute of Marine Engineers in May last. The inherent characteristics of the free-piston gas-generator turbine system are described, and the inward compressing generator is shown to be superior to the outward compressing machine. The disadvantages of the latter are fourfold:—(*a*) It entails a more complicated and expensive machine with heavier reciprocating parts; (*b*) because of the heavier reciprocating parts, the speed of oscillation is lower and thus the maximum output is lower; (*c*) piston cooling is more difficult; and (*d*) the outward compressing type is not so easily maintained.

The duty imposed on the turbine is not by any means arduous, as the maximum inlet temperature seldom exceeds 900 deg. F., while the compressor has proved itself to be as fully reliable as a normal Diesel engine.

The improvements to be obtained by compounding with a steam turbine are examined in the paper, the basic principle being the use, in a steam boiler, of the heat rejected from the gas side, which consists of the gas-turbine exhaust and the jacket loss from the gas generator. Such an installation requires no special materials, and, depending on the actual arrangement, a specific fuel consumption of 0.4 to 0.38 lb. per S.H.P. per hour is expected. This figure could be reduced to 0.35 lb. per S.H.P. per hour, by the addition of a combustion chamber between the gas generator and gas turbine in which additional fuel would be burnt, thus providing a measure of preheat. In this case, a high-temperature gas turbine would have to be provided.

During the past 10 years, six different designs of naval gas turbines have been designed and tested under Admiralty contract, and a further two designs have been taken direct from manufacturers for evaluation and testing. In a paper entitled "British Naval Gas Turbines," read before the Institute of Marine Engineers, Commander (E.) G. F. A. Trewby, R.N., takes each in turn, giving a brief history of development and manufacture. He describes the special features of design and gives details of operating experiences to date, both ashore and afloat, together with the conclusions and lessons to be learned.

Since warships seldom use the full power of their propulsion plant, the life of naval machinery at high powers can be extremely short, judged by commercial or merchant-service standards, and this fact is of great significance when considering the application of gas-turbine machinery to warship propulsion. The majority of time is spent at cruising power, which may lie between 5 per cent, and 30 per cent of the total installed power. An extreme case is the destroyer, which (according to United States naval sources) may spend 99 per cent of its life at 30 per cent power or less.

It is concluded that the greatest single advantage that the gas turbine can offer for warship application is the lightness and compactness, particularly inherent in lightweight open-cycle types.

The experimental gas turbine-driven generator set manufactured by Messrs. W. H. Allen, Sons & Co., Ltd., Bedford, has been so successful that the firm have begun the development of a 350-kW. gas turbine for use on shipboard. This set is practically twice the size of the experimental set. In general layout, it resembles the experimental unit, but a two-stage axial compressor replaces the single-stage

radial, owing to the greater air-mass flow. The reverse-flow combustion chamber is designed to burn any distillate fuel, and is mounted vertically alongside the rotor. The design is such that the vertical height of the machine does not exceed 6 ft. 9 in., and it also permits of ready access to the atomizer. Speed reduction is again by means of an Allen-Stoeckicht epicyclic reduction gear.

The maximum inlet temperature is to be 1,380 deg. F., and all four faces of the two turbine discs are cooled by air tapped from the labyrinth seal. Under these conditions, and with an air-inlet temperature of 86 deg. F., the specific fuel consumption is estimated to be 1.39 lb. per S.H.P. per hour (a thermal efficiency of 10 per cent). This compares favorably with the 1.6 lb. per S.H.P. per hour of the experimental engine, which operated with a turbine-inlet temperature of 1,435 deg. F., but it is still much too high to be tolerated for anything but stand-by use or emergency services.

The same comment concerning fuel consumption also applies to the 60-H.P. portable quick-starting single-shaft gas turbine which the Rover Co., Ltd., of Birmingham, have developed. The single shaft runs at 46,000 r.p.m., and the output speed can be arranged to suit customers' requirements. A prototype fire pump powered by such a unit has been tested, and discharged 400 gallons per minute at a total delivery head of 231 ft. from a static lift of 10 ft. The pump is of a specially designed, single-stage centrifugal type, running at 4,500 r.p.m. The priming is induced by means of an air ejector, which is motivated with compressed air, tapped from the compressor discharge to the combustion chamber. The complete unit (which includes a fuel tank giving a 25-minute duration) is mounted on tubular steel framing and weighs about 200 lb.

REFRIGERATION AND AIR-CONDITIONING

The adoption of Freon as a refrigerant shows a steady increase, and of refrigerated ships built since the Second World War, 50 per cent employ it as the refrigerating medium. Most American ships use Freon, although most British designers still seem to prefer carbon-dioxide for large installations, probably on account of its universal distribution and cheapness.

Apart from these factors, Freon has several important advantages, among which is the comparatively low maximum pressure required (of the order of 120 lb. per sq. in.), even when working in the tropics. This low pressure allows lighter scantlings for the compressor and condenser coils, and lends itself to compression in high-speed multi-cylinder compressors.

The adverse effect of rise of temperature of condenser circulating water is much less marked in the case of Freon, than when using carbon-dioxide as the refrigerant, so that the additional capacity required to meet sea temperatures in excess of 83 deg. F. is much less for Freon than for carbon-dioxide. As the distribution is improved, and the price reduced, the use of Freon for big installations should continue to increase.

A useful chart, summarizing data of refrigerating plants from 41 ships, appears in *European Shipbuilding* (Volume 3, No. 3), and a method is outlined for estimating compressor-cylinder volume for a particular duty.

Recently installed air-conditioning plant in British naval vessels is of the direct-expansion type, acting as a heat pump, so that the sea-water heat-exchangers and the air-conditioning units have to act either as evaporators

or condensers, according to which way the heat is being pumped. Certain submarine plants have a designed capacity of 55,000 B.Th.U. per hour when cooling with condenser circulating water at 90 deg. F. (the sea-water temperature) and the evaporator (chilling water) at 55 deg. F., the heating capacity is 30,000 B.Th.U. per hour from an evaporator circulated with sea-water at 40 deg. F.

H.M. survey ship *Vidal* is designed for long periods of operation in both tropical and polar seas. For purposes of air-conditioning, the ship is divided into three separate sections, each of which is equipped with its own plant. On the cooling cycle, the machinery is designed to operate with a sea-water temperature of 90 deg. F., at a load of 810,000 B.Th.U. per hour, which is sufficient to maintain inside conditions of 85 deg. F. dry-bulb and 73 deg. F. wet-bulb (56 per cent R.H.) when outside conditions are 88 deg. F. dry bulb and 80 deg. F. wet bulb (70 per cent R.H.). When heating, the plant can maintain an inside temperature of 60 deg. F., with an outside temperature of 20 deg. F., which requires a heat load of 750,000 B.Th.U. per hour. In both circumstances, a minimum of 600 cu. ft. of fresh-air per man per hour is allowed for.

Comparison of results of the two types of air-conditioning plant, installed in two heavy cruisers of the United States Navy, has yielded interesting results. One of the vessels used a steam-jet system for producing chilled water for the cooling system, while the other employed a Freon plant with direct-expansion cooling. Despite the increased weight of the chilled-water system, simplicity of its control and the practical absence of maintenance work made it the more favored installation, and it is understood that it will be exclusively adopted on all large-scale installations for later ships.

In order to give the maximum degree of comfort to all persons on board, especially passengers, what may be described as a two-stage air-conditioning system has been devised, and is manufactured in Britain by Messrs. Mechans, Ltd., of Glasgow. It is called the Indivent system. Automatically-controlled central air-conditioning plants pre-treat the air and carry out the primary conditioning made necessary by the climatic changes through which the ship passes. This air is distributed through heat-insulated ducting to the spaces served by the system, and at each supply position, usually in each such space, a secondary means of manual control is fitted. This consists of a booster and air-heater with dial control, which enables rapid adjustments to be made, within certain limits, of both temperature and volume of air finally discharged through the Arnix grill into the room. The air is allowed to exhaust into the adjacent alleyways, except from spaces such as galleys and hospitals, where separate exhaust fans and ducting are provided.

The heating element of the Indivent booster is a hot-water circulated copper coil, and although there is a considerable pressure drop across the unit, it remains virtually noiseless. They are supplied in two sizes for nominal air flows of 150 and 300 cu. ft. per minute.

ATOMIC POWER

A number of American industrial groups were given contracts to study the possible application of atomic power for ship propulsion, with special reference to ships of the "Mariner" class cargo vessel. The prime mover was still to remain as a steam turbine, so that the reactor plant would replace

the oil-fired steam boilers. A summary of six proposals is given in a paper read before the American Institute of Electrical Engineers by Mr. H. L. Witzke and Mr. S. A. Haerstick, entitled "Nuclear-power Plant for Ship Propulsion — Application."

Most reactors have a moderating medium, which serves to slow down the neutrons given out in the fission process to speeds most effective for producing further fission, and to which the heat made available by atomic fission is first transferred. The generated heat is transferred, by means of a coolant circuit, from the moderating medium (or from the core, if no moderator is used) to the steam-raising plant. By using the same fluid for coolant and moderator, the plant can be simplified to a certain extent.

In reactor No. 1 of those quoted in the paper, moderation and cooling is by means of heavy water, natural uranium being the fuel. Reactor No. 2 uses a fuel of slightly enriched uranium, and has light water for moderation and cooling. The third reactor is moderated by heavy water, cooled by light water and uses natural uranium fuel. Graphite is used to moderate reactors Nos. 4 and 5, and liquid sodium for a coolant permits the use of reasonable temperatures without the need for pressurizing the cooling circuit; slightly enriched uranium fuel is proposed. As it is fueled with highly enriched uranium fuel, reactor No. 6 requires no moderation, and sodium is proposed as coolant.

A summary is given of the estimated cost of each proposal, and the conclusion is reached that nuclear-power plants of this type cannot at present compete on an economic basis with conventional power plants. A further serious disadvantage is the considerable weight of the shielding necessary to surround the plant, for the protection of the operating personnel.

Information concerning the two nuclear-power plants being actively developed for submarine propulsion is given in a paper read before the American Institute of Electrical Engineers by Mr. F. E. Crevet and Mr. T. Trocki, and further details concerning one of the plants (S.T.R.) is included by Mr. L. H. Roddis and Mr. J. W. Simpson in their paper presented to the American Society of Naval Architects and Marine Engineers, in November last.

The submarine thermal reactor (S.T.R.) is, it may be stated, already in operation, and the submarine intermediate reactor (S.I.R.) is under construction. Both include the same basic components, and both are to be used as steam generators, the principal source of difference being in the coolant systems. The S.T.R. is cooled with pressurized water, and the S.I.R. is cooled with liquid sodium. The nuclear, thermodynamic and mechanical characteristics of the two systems are discussed, and the points of difference summarized.

Also reported is a preliminary study of a unit using helium as the reactor coolant, the heat being transferred to a closed-cycle gas-turbine plant. Helium does not become radio-active, and thus the heat-transfer system requires no shielding, with consequent reduction of plant weight and improved flexibility of machinery layout. Certain assumptions have been made in this preliminary study which require experimental verification, but it has considerable interest, and much useful information is given in the comparison of the main mechanical characteristics of nuclear plant cooled by water, sodium and helium.

MAN-POWER

In his presidential address to the North-East Coast Institution of Engineers and Shipbuilders, Mr. P. L. Jones, M.C., referred, in particular, to the personnel problem confronting the marine engineering industry in both its operational and manufacturing aspects. The effect of engineering man-power shortage is felt most acutely by shipowners, and the position is more serious now than ever before, because machinery tends to become ever more complex in the search for greater efficiency, while, at the same time, men are increasingly more difficult to attract. Although running costs depend largely on the type and design of machinery, they are also greatly influenced by the ability and quality of the engineers, and there is little point in providing high-efficiency, expensive machinery unless it can be continually run at its peak efficiency and be properly maintained. In the President's own words, "Assuredly, if marine propulsion is to take advantage of future developments, the question of personnel will be one of increasing importance. If it is not soon satisfactorily disposed of, it is quite possible that marine machinery will tend not to advance with the times, but rather to revert to simpler and less economical types."

This problem of attracting and retaining well-trained engineers is being tackled by enlightened firms, and, as the interests of the nation are bound up in the prosperity of the shipping and shipbuilding industries, it is a problem which must be solved in a satisfactory and lasting manner.

MISCELLANEOUS

Corrosion.—

In a series of articles appearing in *Corrosion Prevention* (Vol. 1; April, 1954), Mr. K. A. Spencer and Mr. D. A. Lewis particularly discuss cathodic-protection applications and equipment. They point out that, for vessels laid up and out of service, cathodic protection of the hull is an established practice, and that satisfactory coverage can be readily obtained by placing anodes at points sufficiently distant from the hull to allow of uniform current spread.

However, when protecting active ships by this means, the anode must be attached to the hull itself and there is the consequent danger of local heavy-current flow stripping-off paint. At points close to the anode, the resulting alkalinity and hydrogen evolution can blister the paint and expose bare metal, and, due to the resulting short-circuit, protection at more remote parts is reduced. Thus, while it is essential for the anode to be attached firmly to the hull (though insulated from it), it is equally necessary that the electrically insulating coating of the hull shall be durable for a radius of 5 ft. round the anode. It is suggested that this is best secured by the use of primers, consisting of zinc or lead chromate, in a synthetic-resin carrier, thinning being done by means of an alcoholic solution of phosphoric acid just prior to use. Such primers dry rapidly, and provide satisfactory keys for anti-corrosive paints, good results having been obtained when those with a vinyl base have been used.

For the satisfactory cathodic protection of a surface immersed in sea-water, an initial polarization is said to be necessary. This can be induced by temporary excessive current densities. The level of the final protective potential, hull to water, is quoted as being between −800 and −840 millivolts relative to a silver, silver-chloride half-cell.

Power-impressed anodes are usually of steel or cast-iron, but one of graphite has been developed, which is impregnated with paraffin wax, linseed oil or resin to resist spalling effects due to gassing.

A most promising impressed-current anode is mentioned in an article on "Cathodic Protection of Internal Surfaces," by Mr. H. M. Powell, in the May issue of *Corrosion Prevention*. It is made of platinum mounted on glass in the form of a wire helix, to give maximum surface area for minimum weight of material. High current densities can be used, and long life, coupled with low installation cost, makes an economic proposition for such electrodes. It is also reported that successful tests have been made in the United States of America, using anodes of high silicon iron, but development has not yet been completed.

The following comments are made on various types of sacrificial anodes. Zinc anodes containing 1 per cent of magnesium are being developed, which have more satisfactory anodic-polarization characteristics than commercial zinc anodes, and which do not result in evolution of hydrogen (a particularly dangerous occurrence in ballasted spaces). Zinc has a low-current output per pound, but, since it suffers little wastage, its electrode efficiency is high. Aluminum is subjected to considerable anodic polarization, but it has the highest available current output per pound. On the other hand, magnesium anodes exert the highest potential difference between themselves and steel, and suffer but little anodic polarization.

Because of occasional previous trouble with the corrosion of new ships while being fitted-out afloat, the cable ship *Recorder* was protected by a number of Guardion electrodes suspended at various depths in the water at a distance of 3 to 4 ft. from the hull, while lying alongside a quay at the works of Messrs. Swan, Hunter & Wigham Richardson, Ltd. On dry-docking, there was no sign of attack on the hull.

The natural resistance to corrosion of aluminum has demonstrated itself in the success of an experimental installation of aluminum heating coils, manufactured by Steels Engineering Installations, Ltd., Sunderland, which has now been in service in the center tanks of a British tanker for almost 18 months, and, on recent examination, showed no sign of overall attack on the inside or outside. It is claimed that the overall heat-transfer rate between steam and oil is greater than can be obtained, under similar conditions, with either mild-steel or cast-iron pipes, whether finned or plain. The weight of the comparable mild-steel installation is said to be 2.8 times as great and that of the comparable cast-iron installation, four times as great. The cost is stated to be approximately the same as that of the normal equivalent in mild-steel. The proper insulation of the coils from the ship structure is essential for the success of the installation, and this is achieved by the use of a well-designed, very robust, non-conducting pipe support.

Although lead is strongly resistant to atmospheric corrosion, aluminum is proving itself a superior material for use as sheathing for electrical cables. Its corrosion resistance is equal to that of lead, and its high mechanical strength, good electrical conductivity and greatly superior fatigue resistance make it an attractive proposition. An aluminum sheath presents a reliable earthing connection to stand up well to the rough treatment which can occur during, say, repainting. Furthermore, it is not subject to crystal growth and intercrystalline cracking as a result of vibration, which occasionally has been a serious trouble with lead sheathing. In the event of fire, the comparatively high melting point of alum-

inum (twice that of lead) provides a greater margin of safety.

A novel attempt to combat the corrosion problem in tankers is the proposal to use double plating for the bottom, sides and bulkheads, keeping the frames between the double plating. Mr. D. B. Bannerman presents the case in a paper, entitled "A Structural Approach to the Problem of Corrosion in Tankers," read before the American Society of Naval Architects and Marine Engineers at the annual meeting in November. The weight of such a structure compares favorably with conventional tanker construction, but the advantage of presenting a smooth surface for cleaning and/or protection, although considerable, is partly offset by the loss of carrying capacity.

In engine-cooling systems, and in the absence of chloride, corrosion can, of course, be readily inhibited by small concentrations of chromates. However, in the presence of chlorides, rusting of mild-steel and cast-iron will occur, even in the presence of solutions having 2 to 3 per cent chromate concentrations. Further investigations have been carried out by Fiat Stabilimento Grandi Motori with increased concentrations of inhibitor, and it has been found that a 10 per cent addition of sodium bichromate will protect mild-steel and cast-iron in a hot solution (170 deg. F.) containing 1 per cent sodium chloride. As a result of the investigations, it is suggested that the most convenient inhibitor to use is a 1 to 1 mixture of sodium bichromate-sodium carbonate for cooling-water systems likely to be contaminated with sodium chloride. In an article reporting this work, which appeared in *Gas and Oil Power* (March, 1954), a method of checking both chromate concentration and pH value is suggested.

Overloading of Gears.—

An article by Mr. H. M. Hiersig, published in *Zeitschrift des Vereines Deutsche Ingenieure* (11th March, 1954), examines the design of gears, taking particular account of the effect of periodic loading peaks on fatigue resistance. Analysis of torque oscillograms is used to show that even transient peak loads should not exceed the safe limit for the material. It is pointed out that tooth-loading during starting-up can greatly exceed the steady running load, and this should be catered for in the design.

The duty of any proposed gear is classified into ranges labelled "regular," "intermittent" and "shock" loads. Each range is assigned its appropriate operating constant, which is a factor used to modify the normally required safety factor, and so, in the final design, the normal safety factor is only required to allow for the risks introduced by irregularities in manufacture and material quality. Examples are given of the derivation of such operating constants from information quoted in the literature of the American Gear-manufacturers' Association; there is also extensive tabulated data.

A Dry Lubricant.—

Some properties of molybdenum disulphide as a lubricant are stated in *Polytechnisch Tijdschrift* (7th July, 1954). It can be employed with satisfactory results as a dry lubricant at pressures up to 45 tons per sq. in. and over a temperature range of −60 deg. F. to 750 deg. F. It has a laminar structure resembling that of graphite and exhibits remarkably good adhesion, the coefficient of friction always being low and decreasing with an increase of bearing load. A most effective method of applying this lubricant is in admixture with corn syrup, or it can be employed in either cake or powder form. Oxidation cannot be detected at temperatures below 750 deg. F., but it does occur above this temperature, and increases in rate with temperature rise. The principal disadvantage of oxidation is hardness of the product, molybdenum trioxide, which, consequently,

exerts a scouring action. However, it is shown that, even under oxidizing conditions, the primary lubricating property is retained so long as an underlying layer of unoxidized molybdenum disulphide persists. The chemical stability is good, being unaffected by water, mineral lubricants and most acids, except concentrated hydrochloric acid and *aqua regia*.

When impregnated with molybdenum disulphide, the coefficient of friction of a particular bronze is reduced from 0.3 to 0.14 at room temperature, and at 750 deg. F. the coefficient rises to 0.2. Impregnation reduces the coefficient of friction of many plastics, that of perspex, for example, being reduced from 0.45 to 0.15. It is effective in lubricating parts made of either natural or synthetic rubber; it is an effective inhibitor of fretting corrosion; and it can be used as a sealing compound for screw-threaded joints subjected up to about 1,000 deg. F.

Oil Separation.—

Holford Processes, Ltd., have developed and demonstrated an oily-bilge separation unit enclosed within a vertical drum. Primary separation is carried out in the lower end, where a central perforated injection pipe directs the oily bilge into the field of action of a rotating conical baffle. In its passage through the rest of the apparatus, the mixture is further broken up, and final separation is encouraged by means of a series of baffle plates having staggered holes. An automatic float-operated oil-discharge valve is arranged at the top, and arrangements are made to discharge the effluent water either directly overboard or through a filter chamber. An electric immersion heater or a steam coil can be arranged.

A unit having a capacity of 10-15 tons per hour is 3 ft. 6 in. in diameter, 7 ft. 6 in. in height, and weighs about 17 cwt.; three other sizes are available, the largest of which is rated at 40-50 tons per hour.

In connection with the feed and cooling systems, the Lawson De-Oiler has been installed in several vessels during the past 12 months, and continues to demonstrate its great effectiveness. From emulsions, as well as from heavily contaminated water, the discharge has consistently been found to contain less than one part of oil per million. There is no filter bed, the separating medium consisting of concentric cylinders made of specially treated, knitted, wire-mesh wound on spools. There are no moving parts, and backflushing can be used for cleaning out at suitable intervals. The De-Oiler is made in units 5 ft. in diameter and 4 ft. to 5 ft. in height.

Cargo Handling.—

Many of the self-unloading bulk carriers are fitted with electric drive, in order to take advantage of using the main propulsion generators in a dual capacity, by employing them in port to energize the unloading gear. In this respect, the geared steam-turbine machinery is at a disadvantage, as it is only the main boilers which can be used in the dual rôle, and the port load has to be separately generated.

A new bulk trade has commenced between North Africa and France, in the transport of wine in specially enamelled vats. Three quarters of the trade is now in bulk, the remainder being in casks.

A different kind of conversion has been practised in the *Tina* (177 ft. long), which has recently been converted from an oil-tanker to a bulk butane-carrier. This vessel can carry 400 tons of butane in 14 cylindrical containers, installed vertically and tested to 250 lb. per sq. in. The possibility of transporting methane or natural gas from the oil fields in the same manner is also receiving serious consideration.

A very successful electric winch de-

veloped by Nordseewerke Emden G.m.b.H. incorporates a luffing winch, which can be driven from the main motor through a claw coupling. The motor develops 25 H.P., and is fully protected against sea-water. The inclination of the derrick can be altered by means of the luffing winch under loads up to 1½ tons, and the gear ratio between luffing winch and main drum on the cargo winch is such that, during the luffing operation, the path of the hook is practically horizontal. The winch makes especially quick work of hatch covers.

Hatch covers of the type constructed of hinged sections, which fold up when opened, can now be hydraulically opened and closed by means of a self-contained system energized by patented hydraulic accumulators having a capacity sufficient for two openings and closings before the accumulators need be recharged. A small, 1-H.P., electric motor-driven pump recharges the hydraulic accumulators. Opening and closing is the work of a few minutes for one man.

The overall record of the "Mariner" ships in service, up to the 1st April, 1954, has been summarized in a paper, entitled "Operation in Service of the 'Mariner'-type Ship," read before the American Society of Naval Architects and Marine Engineers, in November last, by Mr. W. G. Allen and Mr. E. K. Sullivan. Apart from the deck and engine-room log abstracts normally used by shipowning companies, a supplementary sheet was devised to collect, for analysis, information which is not normally noted. The extra data concerned condition of loading, sea-keeping characteristics, opening of hatches, rigging and stowing cargo-handling gear, and rate of loading and discharging cargo. Along with these records, considerable cost data were collected and are included in the paper. The information is itemized in great detail, and throws considerable light on to the many ways in which economies can be made.

There is a saving attributable to the use of folding hatch covers, which substantiates their continued employment, though a greater saving is visualized by divorcing the cargo-handling gear from the opening and closing operations. A comparison of the "Mariner" cargo gear with the Ebel-type gear installed in the *Schuyler Otis Bland* indicates the clear superiority of the latter type.

The difficulty of designing a ship to satisfy the requirements of both military and merchant shipping interests is demonstrated by the light draft under which the "Mariners" have sailed, even when fully laden. The designed draft for merchant-service use is 29 ft. 10 in., while the average draft, when loaded with military cargo, has been 22 ft. 6 in.

Vapor Compression Evaporators.—Several references have appeared in the technical Press on the use of vapor compression evaporators, and they appear to be a very attractive proposition. The boiling process can be started in the main evaporator chamber by some suitable means, such as an electric immersion heater. Once boiling is well started, the issuing vapor is compressed mechanically in a rotary compressor and passed back through a coil immersed in the boiling fluid. Due to compression, the temperature and pressure of the vapor are raised, and, on its passage through the coils, a certain quantity of heat is transferred to the bulk of the fluid in the evaporator, resulting in condensation of part of the compressed vapor and the formation of almost the same weight of vapor within the evaporator. The condensate thus formed is the output of the evaporator, the remaining steam and newly formed vapor being re-cycled.

By this means, the weight of distillate obtained is from 10 to 15 times that produced by a single effect per unit

of heat input. On trials, using a Diesel-driven compressor, up to 275 lb. of fresh-water has been produced per 1 lb. of Diesel oil. The stills can be made in sizes ranging in capacity from 20 gallons to 100,000 gallons of fresh-water per day.

New Edition of Lloyd's Rules.—

The unification of the Rules of the British Corporation Register of Shipping and Aircraft with those of Lloyd's Register of Shipping has now been completed and issued as a new edition of the Rules and Regulations for the Construction and Classification of Steel Ships. The alterations include certain requirements for the materials and thicknesses of piping, and a new formula is used to control the thickness of seamless pipes. For ships being built under special survey, an addition has been made to the machinery items which must be surveyed during construction, and these include air compressors, scavenge blowers, superchargers and fuel-valve cooling pumps for heavy-oil engines.

An important amendment concerning shafting reads as follows:—"Where serious critical speeds are found by calculation to occur within the range of working speeds, the Committee may require torsiograph records to be taken from the machinery for the purpose of verifying the calculations, and may impose a restriction on continuous running at speed where the stresses are considered to be excessive."

On the electrical side, there is a restriction in voltages allowed for use in tankers carrying oil in bulk which has a flash-point below 150 deg. F. For such vessels, using A.C., the supply for power must not exceed 250 volts single-phase, or 440 volts three-phase; while for lighting and fans in accommodation, the limiting figure is 105 volts.

WELDING TYPE 347 STAINLESS STEEL FOR 1100° F TURBINE OPERATION

ACKNOWLEDGEMENT

This article appeared in the March 1955 edition of *The Welding Journal*. The authors are R. M. Curran and A. W. Rankin, of General Electric Company.

Between November 1948 and February 1952, ten large steam turbines totaling approximately 1,000,000 kw capacity were placed in operation at the 1050° F temperature level by the authors' company. All these units utilized austenitic steels of the 316, 321 and 347 families for the major turbine parts such as the inner shells, the stop and control valve casings and heads and the piping between the stop valves and the turbine proper. All the welds in the latter piping were made with Type 347 electrodes with approximately 8 to 10% ferrite content and with no preheat; on the first such unit, most of the piping welds received a 1600° F postweld treatment, but a number of the piping welds on that unit together with all the piping welds on the nine subsequent units received no heat treatment after welding. Although the development of a 1% chromium, 1% molybdenum and ¼% vanadium steel in 1950 resulted in subsequent units for the 1050° F temperature level being constructed of this low-alloy ferritic steel, the operating experience gained on the ten units constructed with the austenitic steels was of invaluable assistance when design work was started on the first 1100° F unit—the Kearny No. 7 unit of the Public Service Electric and Gas Company of New Jersey. On this unit, the step to the higher temperature level resulted in the reintroduction of the austenitic stainless steels for the major turbine parts as well as its use for the main station piping.

While essentially satisfactory operation has been obtained with the ten 1050° F units employing the austenitic stainless steels, cracking has occurred in and adjacent to the welds to a relatively greater degree than has been experienced with the ferritic steels. Such cracking has been caused either singly or by a combination of a greater notch sensitivity of the austenitic steels, a slower relaxation of residual welding stresses, or excessive thermal-expansion stresses. In addition, examination of a number of these welds, together with long-time aging tests, have shown that their ductility decreases with time at the operating temperature due to the formation of sigma phase in the deposit itself. While the effects of notch sensitivity residual

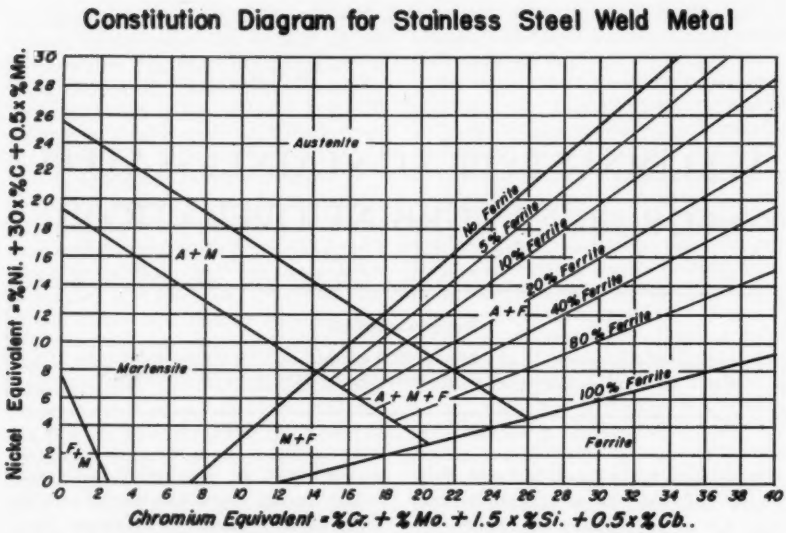

Figure 1. Constitution diagram for stainless steel weld metal after Anton L. Schaeffler showing the effect of weld metal composition on its strucure.

welding stresses and thermal expansion stresses can be minimized, and perhaps eliminated, by design and erection modifications, the elimination of the sigma formation appeared to be a more difficult problem, and, in particular, appeared to be one whose effects would become more pronounced as operating temperatures reached and exceeded the 1100° F level. Accordingly, a series of technical meetings were held in late 1951, sponsored by the Public Service Electric and Gas Company of New Jersey, to discuss welding electrodes and procedures through which sigma phase formation could be minimized in the austenitic welds to be used in the projected 1100° F units. Investigations being conducted at that time and bearing on this subject by some of the participants in this series of technical meetings have been presented in prior publications.[1-3] The purpose of this report is to describe the investigations conducted by the authors' company in this development of an electrode and welding procedure through which Type 347 welds could be obtained which would be essentially insensitive to sigma phase embrittlement during service at 1100° F and higher temperatures.

THEORETICAL CONSIDERATIONS

In the early efforts to weld the austenitic stainless steels, considerable weld-bead and crater cracking was encountered. Studies of the effect of weld metal composition disclosed that higher chromium and lower nickel

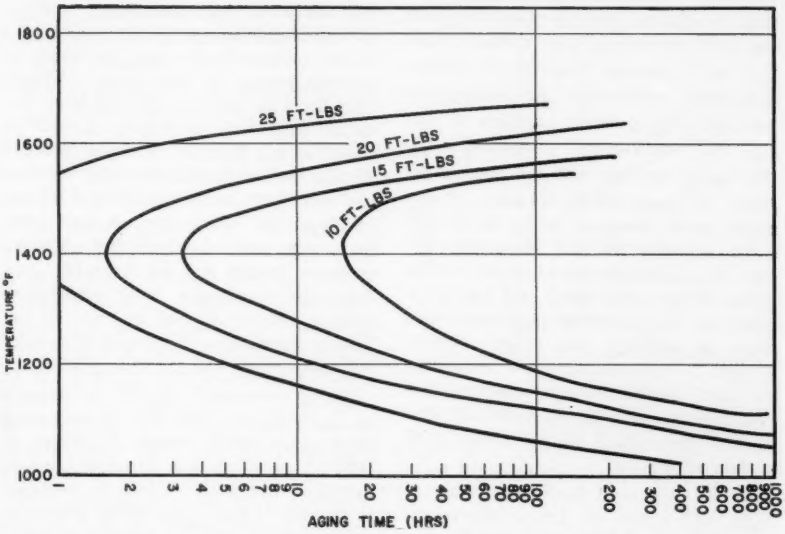

Figure 2. Effect of aging time and temperature on the room-temperature keyhole-Charpy impact strength of Type 347 weld deposits containing 5-10% ferrite. Initial impact strength 30 ft-lb.

contents in the deposited weld metal produced a greater resistance to this type of cracking. Metallographic studies of such higher chrome—lower nickel weld deposits showed that they contained small percentages of ferrite in the austenitic matrix. The effect of changes in weld-metal composition on the percentage of such ferrite in weld deposits is shown in Fig. 1.[4] With the equivalent chromium and nickel contents of the customary Type 347 weld deposits, an increase in the chromium content or in the content of elements that act like chromium or corresponding decreases in the nickel content, tend to increase the percentage of ferrite present in the weld deposit.

While the immunity of such weld deposits to cracking during welding has been accepted for some time, and weld metal compositions of Type 347 have been designed to contain percentages of ferrite, the effectiveness of ferrite in preventing cracking still is not completely understood. The theory has been advanced that the cracking of fully austenitic weld deposits is due to the formation of low-melting silicate films in the columnar grain boundaries,[5, 6] and that these films rupture upon shrinkage of the bead immediately after solidification. The effectiveness of the presence of ferrite in preventing this type of cracking has been attributed to the fact that the silicate films probably form around the ferrite particles instead of in the grain boundaries, and hence are made ineffective in so far as cracking is concerned.

Although the ferrite content required to eliminate weld-bead cracking probably is very small, it is neces-

sary to guard against dilution from the fully austenitic base material into the weld deposit making the latter also fully austenitic; accordingly, it has been the general practice to design the weld-metal composition so that some ferrite will be present in spite of base-metal dilution. Inasmuch as the presence of ferrite in the weld deposits did not adversely affect the moderate-temperature properties of the weld metal and had little effect on the corrosion resistance, and since, in addition, the effect of prolonged high-temperature exposure was not understood, few intensive investigations were conducted with regard to depositing sound welds with no ferrite present.

In the postweld heat treatment of Type 347 welds at temperatures in the range from 1200 to 1500° F,[7] however, it was found that the hardness of the weld metal increased and the ductility decreased due to the conversion of the ferrite to sigma phase. Considerable cracking in the weld and in the base metal immediately adjacent to the weld was also encountered during the postweld heat treatment. Since the postweld heat treatment resulted in a decrease in ductility as compared to an increase in ductility for welds in ferritic steels, it was quite obvious that the sole benefit derived from the usual postweld treatment would be relief of the residual stresses induced by the welding operation. In view of the excellent ductility of weldments in Type 347 as-welded, it had been generally agreed that the residual stresses would not be harmful, and that the postweld heat treatment should not be applied in view of the loss in ductility and danger of cracking associated with such a procedure.

Aging tests conducted on weldments of Type 347 at 1100° F and operating experience with steam turbines at 1050° F showed that Type 347 weldments would suffer a loss in ductility in service at these operating temperatures. It thus was apparent that elimination of the postweld heat treatment merely postponed the time at which this loss of ductility would occur. An investigation was initiated, therefore, to determine the extent to which the loss in ductility would proceed and, if possible, to determine means of preventing such embrittlement.

ELECTRODE AND HEAT-TREATMENT DEVELOPMENT

Although material properties such as tensile strength, yield strength, rupture strength, creep strength and the like are directly convertible to allowable design limits, it is difficult in any type of construction to decide upon what basis satisfactory ductility limits can be set. In the absence of suitable criteria to establish such limits, every effort is made to provide as much ductility as possible without undue sacrifice of tensile properties, and an evaluation of the ductility levels achieved is made on the basis of the suitability of similar materials operating under similar circumstances.

Since the Charpy impact test has been shown to be the most sensitive measure of sigma embrittlement, it was decided, in this investigation, to use the room-temperature Charpy impact strength as a measure of the extent of embrittlement of welds at elevated temperature. Although the results of such tests are not directly convertible to reduction of area or percent elongation, it was assumed, for the initial investigations, that an improvement in the room-temperature

Charpy value corresponds to an improvement in the operating ductility of the weldment.

The first step in the investigation was a study of the effect of aging temperature and time on the Charpy impact strength of Type 347 weld deposits. The resulting data are plotted in Fig. 2 showing the time at temperature required for the weld deposit to drop to the indicated level of impact strength from the value of 30 ft-lb obtained in the as-welded condition. This figure is only schematic because of the limited data and because it is peculiar to each particular composition as changes in the ferrite content would undoubtedly shift the figure on both the horizontal and vertical axes.

It appeared probable from examination of these and other data presented in the literature that the reduction of ductility caused by sigma phase formation is due to the formation of this phase, with essentially no ductility, in an austenitic matrix whose properties are unaffected by the local conversion of ferrite to sigma phase. It seemed reasonable to expect, therefore, that the reduction in impact strength would be a function of the percentage of the fracture path which occurs through sigma phase, and that any steps which would reduce this percentage would also improve ductility. This resolved itself then into the problem of both reducing to a minimum the volume of ferrite and spheroidization of the remainder. Since the 1650° F treatment was found to both spheroidize the ferrite and reduce its volume, it appeared that by utilizing a low initial ferrite content together with a suitable heat treatment, a minimum of ferrite would be present to convert to sigma, and would be essentially spheroidal in form so that its conversion to sigma would not cause any serious loss in ductility.

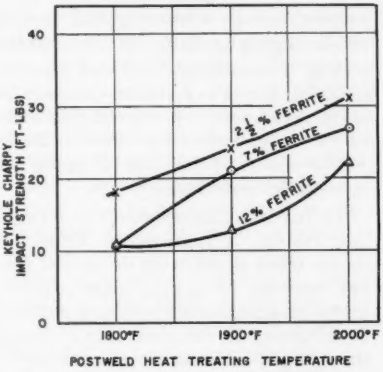

Figure 3. The effect of initial ferrite content and postweld heat treatment on the impact strength of Type 347 stainless steel weld deposits after exposure at 1350° F for 300 hr.

In order to test the foregoing hypothesis, a number of Type 347 welds were made with initial ferrite contents ranging from 2½ to 12%, with postweld treatments varying from 1800 to 2000° F, and with subsequent aging at 1350° F for 300 hr. The latter aging was intended to simulate the effect of long-time exposure at the lower operating temperatures, and the 1350° F was selected from the data of Fig. 2 which indicated this to be close to the temperature of most rapid sigma conversion. The keyhole Charpy impact strength of all these welds was approximately 30 ft-lb after the postweld treatment but before the 1350° F aging treatment, and the effect of the latter aging treatment on the impact strength is presented in Fig. 3. As a basis to evaluate the efficacy of the lower ferrite contents in conjunction with the high-temperature postweld treatments, a Type 347 weld with an initial ferrite content of approximately 8% and with no postweld treatment would drop in impact strength from an initial value of 30 ft-lb to approxi-

mately 7 ft-lb when exposed to this 300-hr aging at 1350° F. The results of Fig. 3 accordingly indicate that by utilizing the lower ferrite contents in conjunction with a postweld treatment in the neighborhood of 1900 to 2000° F, the effects of the 1350° F aging are minimized considerably.

To further substantiate the foregoing results, a number of Type 347 welds were made with deposited ferrite content of approximately 2½%, and with a postweld treatment at 1925° F. In order to simulate more closely the aging effects of many years of service at the operating temperature of 1100° F., the welds of this series were aged at 1200° F for periods ranging out to 6400 hr. The results of these tests are shown on Fig. 4, on which are also shown the effect of 1200° F aging on Type 347 welds with between 5 and 10% initial ferrite content aged in the as-welded condition (no postweld treatment). Figure 4 agrees with Fig. 3 in showing that by utilizing a low-ferrite electrode in conjunction with a high-temperature postweld treatment, a marked improvement is obtained in the resistance to service embrittlement.

While the foregoing tests demonstrated a means of minimizing service embrittlement, the use of such a low-ferrite electrode involves the risk of cracking during welding if base metal dilution causes the weld deposit to become fully austenitic. Previous publications[5,6] have demonstrated, however, that sound welds can be made in the fully austenitic 25% chromium—20% nickel and 15% chromium—35% nickel compositions by properly balancing the silicon and carbon contents. Accordingly, tests were conducted on three series of fully auste-

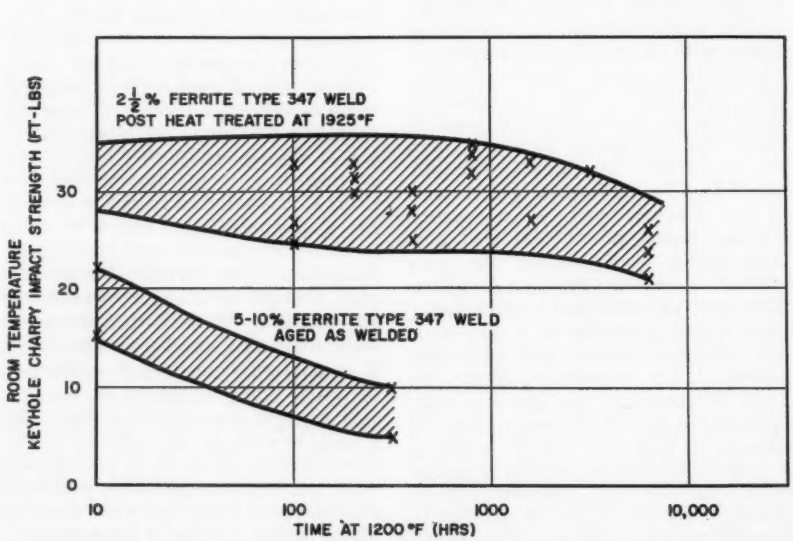

Figure 4. A comparison beween the resistance of postweld heat treated low ferrite and as-welded 5-10% ferrite Type 347 welds to loss in room-temperature impact strength after aging at 1200° F.

nitic Type 347 welds with varying silicon contents and silicon-to-carbon ratios, and sound welds were obtained on such fully austenitic welds provided the silicon/carbon ratio was 5 or less.

In the 25-20 and 15-35 compositions, and in the foregoing fully austenitic 347 welds, the silicon/carbon ratio was maintained at the desired low level by permitting carbon contents higher than in the standard 347 electrodes. Aging tests on the higher-carbon, fully austenitic 347 welds, however, indicated that the higher carbon content resulted in increased embrittlement tendencies, and that it would be desirable, therefore, to maintain the desired low silicon/carbon ratio by holding to a low silicon level rather than by utilizing a high carbon content. Since it would be impracticable, however, to attempt to maintain a very low silicon content in the relatively large quantities of this electrode which would be needed, it was necessary to compromise somewhat on both silicon and carbon by permitting the silicon to go to 0.50% maximum, and by increasing the carbon maximum to 0.10% from its more usual value of 0.08%. The higher carbon content was considered acceptable for power station steam piping since the atmosphere is not particularly corrosive. The columbium content was specified as 0.60 to 0.90% rather than the more usual columbium specification of ten times the carbon content because the contribution of this element in high-temperature applications is in its improvement of rupture strength rather than in its ability to stabilize carbides to resist corrosion. The final specification for the low-ferrite electrode is given in Table 1.

TABLE 1—*Specification Limits for Special Low-Ferrite, Type 347 Welding Electrodes*

Chemical composition, % (deposited weld metal)	
Carbon	0.07-0.10
Manganese	1.50-2.50
Phosphorus, max.	0.025
Sulfur, max.	0.025
Silicon, max.	0.50
Chromium	18.5-20.5
Nickel	9.0-10.5
Columbium	0.60-0.90
Silicon/carbon ratio, max.	5/1
Ferrite content*	1-4

*Calculated from chromium-nickel equivalents.

PROCEDURE QUALIFICATION TESTS

Qualification tests were conducted on welds made in Type 347 steel using the low-ferrite electrode of Table 1 together with the 1925° F postweld treatment. The results are presented in Table 2; the values obtained in both tensile tests and bend tests are unusually satisfactory, particularly in both tensile and bending ductility.

HIGH-TEMPERATURE RUPTURE STRENGTH

High-temperature rupture tests were conducted on these low-ferrite welds with a 1925° F postweld treatment using the Larson-Miller[8] method of trading exposure time for temperature. The results are presented in Fig. 5 on which is also plotted the test data on the high-temperature rupture strength of Type 347 base material.[9] It is evident from this Fig. 5 that the high-temperature rupture strength of the low-ferrite weld deposit is equal to that of the Type 347 base material.

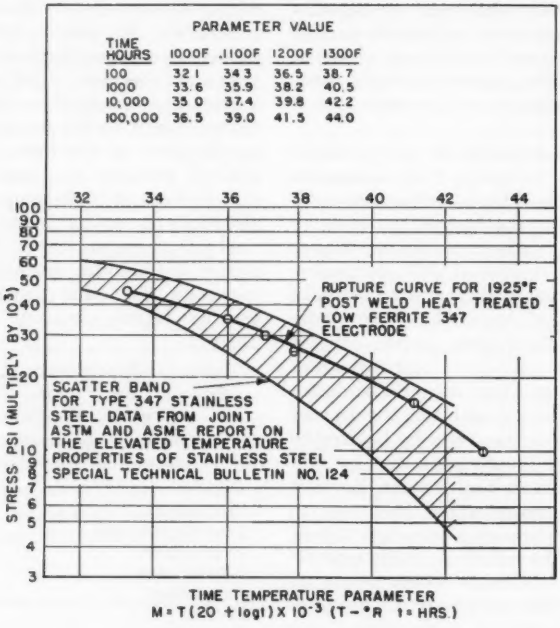

Figure 5. A comparison of the rupture strength of low-ferrite Type 347 stainless steel weld deposits heat treated at 1925° F after welding with the rupture strength of Type 347 base metal. Time (hr) and temperature (° F) plotted on a parameter basis.

POSTWELD-TREATMENT CRACKING

While the foregoing investigation indicated that sound welds could be obtained with a low-ferrite electrode with balanced silicon and carbon, and that excellent resistance to service embrittlement could be obtained in such welds when given a 1925° F postweld treatment, the possibility of cracking during this postweld treatment still remained, particularly in the heavy sections to be welded. Quantitative data on such cracking are somewhat limited in scope and applicability because of the difficulty of simulating weld-joint stresses in a simple test amenable to quantitative analysis. It has been observed, however, that cracking of austenitic welds during the postweld heat treatment is

TABLE 2—*Qualification Test Results*

(*a*) Transverse tensile tests		
Ultimate strength, psi	79,000	79,400
Reductionof area, %	62.6	66.5
Location of fracture	Base Metal	Base Metal
(*b*) Free-bend tests		
Elongation, %	55.0	58.5
(*c*) Side-bend tests	*	*

*All four specimens satisfactory.

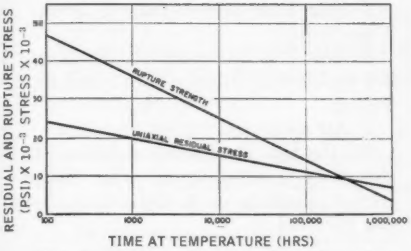

Fig. 6. Curves showing the relaxation of uniaxial residual stress and decrease in rupture strength of Type 347 stainless steel with time at 1100° F.

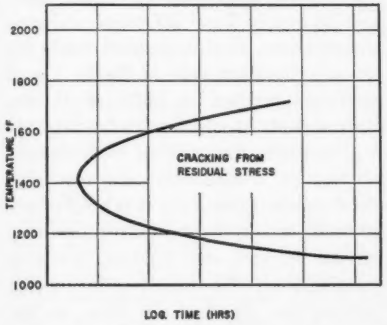

Figure 7. Hypothetical curve showing time at temperature necessary for residual stresses to cause cracking of Type 347 weldments during postweld heat treatment.

accentuated rather than alleviated by slow rates of heating, and that such cracking is generally intergranular in nature indicating failure at high temperatures. These observations indicate that the cracking is due to the residual welding stresses rather than stresses caused by uneven heating.

The residual stresses present in a highly restrained weld may be expected to approach the level of the room-temperature yield strength, and this level will usually exceed the short-time rupture strength at the stress-relieving temperature. Accordingly, there is a possibility of rupture failure occurring during the stress-relieving treatment, although the realization of this possibility depends on the relative rates of decrease of residual stress and rupture strength during the postweld treatment. While quantitative data on these rates are not available for the complex residual-stress pattern present in welded joints, some data are available from uniaxial relaxation[10] creep and rupture tests, and the data from such tests at 1100° F on Type 347 steel are presented in Fig. 6. This figure indicates that, at 1100° F, uniaxial residual stresses of the initial magnitude used in these tests would eventually result in rupture failure; residual stresses in restrained welds, however, would tend to relax at a slower rate because of the effect of multiaxiality, and accordingly failure would be expected at a somewhat shorter time. At temperatures higher or lower than 1100° F, the point of intersection of the residual stress and rupture strength curves would depend, of course, on the rates of decrease of these two factors. Although quantitative data are not available defining this point of intersection for temperatures above 1100° F, the observation that slower heating rates accentuate the cracking of Type 347 weldments during the postweld treatment suggests that the point of intersection of the residual stress and rupture strength curves occurs at a minimum time at some particular temperature. It can be hypothesized, therefore, that a plot of cracking time vs. temperature would be of the general shape shown in Fig. 7, and this in turn implies that the heating above approximately 1200° F should be as rapid as is practicable without introducing excessive stresses due to thermal differences.

On the basis of the foregoing hypothesis, several highly restrained Type 347 weldments were subjected to the following postweld treatment:

1. Heat to 1100° F at 300° F per hour.
2. Hold at 1100° F for 2 hr to permit temperature equalization and relief of the higher residual stresses.
3. Heat from 1100 to 1925° F at 600° F per hour.
4. Hold 2 hr for complete stress relief and ferrite solution and spheroidization.
5. Air cool.

All the test weldments subjected to this heat-treating cycle were found free of cracking upon inspection after the air cool.

SHOP AND FIELD FABRICATION EXPERIENCE

The low-ferrite welding electrode and 1925° F postweld treatment described in this paper have already been used for the fabrication of turbine and piping components in several power-generation stations with satisfactory results. Some difficulties have been experienced with both electrode and postweld treatment cracking, but these difficulties have not been of major consequence. The difficulties with the electrode involved some cracking during welding which is not too surprising in view of the low ferrite level which is being maintained. The nature of this cracking indicated that further study is needed concerning the dilution effects of the base metal, and also indicated that detailed attention must be given to the relation between base metal composition and electrode chemistry. The cracking which has occurred during the postweld treatment has also been minor in nature and easily repaired. In particular, these difficulties with both electrode and postweld-treatment cracking have indicated the need for still more intensive investigation of the characteristics of Type 347 welds, but have not been of sufficient magnitude to cast serious doubts on the practicability of the suggested procedure.

An illustration of the type of components on which this welding procedure has been used is a 20-ft-long stop and control-valve assembly, which consists of forged Type 347 valve and pipe components. It was welded with the low-ferrite electrode of Table 1 and postweld treated at 1925° F in the Schenectady steel foundry heat-treating furnace. Air cooling was accomplished by withdrawing the car from the furnace. Distortion was held to acceptable values by suitable supporting on the furnace car. Surface cracking occurred on the heavy fillet welds joining the small drain lines to the valve bodies, but no cracking occurred on any of the main structural welds. This assembly was subjected to a second 1925° F postweld treatment following the welding of stubs to the six control valve outlets, and no cracking occurred on any major structural welds during this second furnace heat treatment. A second such assembly was also welded and heat treated in a similar manner with no serious difficulties.

The field welds connecting these valve assemblies to their respective turbines, together with all the welds in the Type 347 main steam and turbine piping, were also welded with this low-ferrite electrode and postweld treated at 1925° F using induction coils with no unusual difficulties.

CONCLUSIONS

On the basis of the results presented in this paper, the following conclusions may be drawn:

1. Sound welds can be produced in Type 347 stainless steel using an electrode with a ferrite content in the range of 1 to 4% and with the chemical composition balanced in accordance with Table 1.

2. The use of a 1925° F postweld treatment on welds made with such electrodes results in an appreciable increase in their resistance to embrittlement caused by long-time exposure to elevated temperature.

3. Highly restrained austenitic weldments can be heat treated successfully in the range of 1900 to 2000° F provided a rapid heating rate is used.

4. The long-time, high-temperature rupture strength of low-ferrite welds heat treated at 1925° F is comparable with the corresponding long-time, high-temperature strength of Type 347 base material and Type 347 welds made with the usual commercial compositions.

This paper would not be complete, however, without emphasizing that the authors look on this investigation as only another step in the development of an electrode and welding procedure from which sound reliable welds can be obtained in the austenitic stainless steels. Specifically, this step was intended only to cope with the problem of sigma embrittlement, and even in this isolated area the proof of success must await the results of years of service at the 1100° F and higher temperature levels. In addition, fundamental studies are needed to explain more quantitatively the causes of cracking during the postweld treatment. That which has occurred during the shop and field applications of the electrode and postweld treatment described in this paper has been acceptable on the basis of its being a low price to pay for elimination of sigma formation, but production schedules demand that such cracking be further minimized. In addition, further studies are needed to more adequately explain the service cracking which has occurred in some of the austenitic steel weldments operating at 1050° F as discussed in the Introduction. While it has not been possible to ascribe these to any single cause, there is considerable reason to believe that the complete relief of residual welding stresses afforded by the high-temperature postweld treatment together with the minimizing of the sigma embrittlement will appreciably reduce the incidence of this cracking.

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ENGINEERING ABSTRACTS

Compiled by

JOHN W. SAWYER

Abstract Number	*Abstract Title*
1-55	Tests on the Turbocharged Sulzer Engine
2-55	Pitch Errors of the Serrations of Gas Turbine Blades
3-55	Importance of Engineering Science on the Future of the Profession
4-55	Ultrasonic Attenuation Measurements for Study of the Engineering Properties of Materials
5-55	The Effect of Grinding on the Fatigue Strength of Steels
6-55	Cetane Improver Additives for Diesel Fuels.
7-55	The Comparison of the Fatigue Characteristics of New and Used Crankshafts
8-55	Shipboard Vibrations
9-55	The Development of a High-Speed, Lightweight Opposed-Piston Engine for Submarine Application
10-55	Analysis of the Basic Noise Sources in the Diesel Engine
11-55	The Preparation of Residual Fuel for Motive Power
12-55	Diesel-Engine Piston and Cover Failure
13-55	Lost Propellers
14-55	Fuel Features Related to Operating Experiences in Motor Ships Using Low Cost Fuels
15-55	Cermets—A New Solution for Thermal Shock
16-55	Some Operating Experience at 950°F.
17-55	The Future of the Marine Gas Turbine
18-55	Enclosures for Large Induction Motors
19-55	Crash Resistant Fuel Tanks Are on the Market
20-55	Hysteresis Devices: Torque Independent of Speed
21-55	Dropwise Condensation on a Metal Surface
22-55	Oil Film Extent in Complete Journal Bearings
23-55	Gas Turbine Progress
24-55	Non-Metallic New Materials in Engineering
25-55	Practical Approach to Some Vibration and Machinery Problems in Ships
26-55	A Comparison of Gas Turbines, Weight and Bulk as Factors in Choosing Propelling Plant for Naval Vessel
27-55	The Development of a 500 HP Multipurpose Gas Turbine Engine

1-55 "TESTS ON THE TURBOCHARGED SULZER ENGINE."—"The Motor Ship." v. XXXV. n. 420, March 1955, p. 525. A new R.S. model, 10 cylinder, unsupercharged, 9,000 b.h.p. engine has been operated for 3,000 hours without trouble by Sulzer Brothers at Winterthur, Switzerland. This engine has the lower part of the cylinder separated from the crankcase by a diaphragm. The piston rod passes through a stuffing box in this diaphram. This feature of separating the cylinder and crankcase was to permit

operation with boiler oil. A turbocharged Sulzer engine, seven cylinder, new type, 8,400 b.h.p., completed in Japan, was to be tested in March. A 12 cylinder, 15,000 b.h.p. engine has been started at Winterthur. A twelve cylinder engine is to be produced for a Norwegian concern and will be the highest powered Sulzer engine constructed. Trials on a single cylinder show specific consumption below that of unsupercharged. A specific fuel rate of 150 gr. per b.h.p. hr. is predicted. Tests show exhaust temperature 680°F., firing pressure, maximum, 850 p.s.i. Experimental unit ran continuously for three weeks with a b.m.e.p. of 100 psi.

2-55 "Pitch Errors of the Serrations of Gas Turbine Blades."—L. W. Nicklos. "The Engineer." v. 199. n 5182, May 20, 1955, pp 692-694. A comparator has been developed by The National Physical Laboratory to measure the pitch of the pressure flanks of the serrations of the fir tree roots of gas turbine blades. It can be adapted to measure the pitch of the centers of the grooves. Accuracy requirements are such that measurements within plus or minus 0.00005 in. are desired. Speed is also essential to inspect blades in quantity production. Instrument construction and operation described. Photographs, drawings and references.

3-55 "Importance of Engineering Science on the Future of the Profession." —S. C. Hollister. "Applied Mechanics Reviews," May, 1955. Shortage of engineering manpower is discussed. Need to widen the scientific knowledge of the engineer, strengthen analytical tools, encourage exercise of broad judgment and making of practical decisions is emphasized.

4-55 "Ultrasonic Attenuation Measurements for Study of the Engineering Properties of Materials."—Rohn Truell. Paper No. 55-S-17 presented at the ASME Diamond Jubilee Spring Meeting, Baltimore, Md., April 18-21, 1955. Study of physical properties of metals by a new method is discussed. The method described involves the measurement or the comparison of ultrasonic energy losses in solids. Technique of measuring attenuation, measuring low attenuation values, attenuation of copper-aluminum alloy and titanium-hydrogen experiment are discussed.

5-55 "The Effect of Grinding on the Fatigue Strength of Steels."—D. N. Cledwyn-Davies. "Proceedings of the Institution of Mechanical Engineers," v. 169, n. 2, 1955, pp 83-92. An investigation was made to fix the effects of cylindrical grinding on the fatigue strengths of several steels including: nickel-chromium-molybdenum alloy hardened and tempered to give a nominal ultimate stress of 80 tons p.s.i., the same steel treated to give a nominal ultimate of 62 tons p.s.i., a 0.6 per cent carbon steel 45 tons p.s.i., and a 0.1 per cent carbon steel 27 tons p.s.i. Materials, heat treatment, testing machines, test specimens, grinding fixtures, grinding specimen profile, grinding speeds, wheels, measurement of surface finish, accuracy of testing and roughness are discussed. Photographs, tables, S-N diagrams, and Talysurf recordings of surfaces are included. Conclusions include: (1) Grinding, under proper conditions has no deleterious effect on fatigue strength of the steel. (2) Poor grinding causes reduction in fatigue strength. (3) Stress relieving the ground specimens

of Vibrac V 30, which showed decrease in fatigue strength, failed to restore fatigue strength of ground specimens to that of virgin metal. (4) Reduction in fatigue strength, due to grinding, was due to patches of grinding burn on the surface. (5) Small differences in ground surface finish (2-11 micro-inches) had no apparent effect on fatigue strength of a hardened and tempered steel (Vibrac V 30).

6-55 "Cetane Improver Additives for Diesel Fuels." W. H. Hubner. Paper No. 55-OGP-12, presented at the ASME Oil & Gas Power Conference, Washington, D. C., June 5-10, 1955. Production of diesel fuel is about six times the 1941 usage while gasoline usage has increased two fold. Straight run distillates have a wide range of cetane numbers varying between 33 to 68. Ignition quality of fuel varies with heat of compression. Ignition delay period, from start of injection to beginning of combustion, depends on engine design, operating conditions, atmospheric conditions and cetane rating of the fuel. Higher cetane fuels ignite easier and give shorter delay period than lower cetane fuels. Cetane number is of particular importance for cold starting and low load operation with low combustion chamber temperatures. An increase in each cetane number reduces starting temperature about 1°F. Lowering cetane number will increase the delay period at low load operation. Higher cetane fuels operate on lower temperatures before misfiring is encountered. Engine cetane requirements vary over wide range. Generally a 45 minimum cetane fuel is required. A cetane-improver additive has been developed. This is DB-36 amyl nitrate. It is being used successfully by a number of refiners and jobbers to: upgrade 100 per cent catalytic-cracked distillates for so-called railroad economy diesel fuels; upgrade No. 1 furnace oil and stove oil to bus and premium grade diesel quality; upgrade conventional No. 2 furnace oil to regular-grade diesel quality. The DB-36 additive has been given extensive engine testing in the laboratory, diesel trucks, U. S. Navy submarines and railroad diesels. Characteristics described include: volatility, non-flammability, insensitivity to thermal shock, solubility and stability.

7-55 "The Comparison of the Fatigue Characteristics of New and Used Crankshafts."—J. L. Ciringione. Paper No. 55-OGP-5, presented at the ASME Oil & Gas Power Conference, Washington, D. C., June 5-10, 1955. In a program to develop data on permissible limits of torsional, bending and longitudinal vibration stresses of crankshafts from diesel engines numerous expensive full size elements are needed. Also studies to investigate influence of stress history on endurance limit and S-N relationship of old crankshafts relative to reconditioning plans necessitate many units. A program was undertaken to compare fatigue characteristics of used crankshafts, with unknown history, with new units. Twenty resonance-type fatigue testing machines were built to subject crankshafts to pure stresses in torsion and bending. Tests consisted of subjecting each specimen to each of at least four nominal stress levels of the crankpin. Failure occurred in every case at the intersection of the crankpin surface and the oil hole in torsional test. Failure in bending occurred at

crankpin fillet. All tests were continued to failure or to 30×10^6 reversals of stress. Tests show:

Crankshafts	*Endurance Limits*	
	Torsion	*Bending*
New	6000	7200
Used	7100	7000

It is concluded that new and used crankshafts are equal with respect to fatigue. Used crankshafts show greater endurance limit probably as result of cold-working in operation. Bibliography, illustrations, curves.

8-55 "SHIPBOARD VIBRATION."—Milton J. Berg. Special lecture presented at the ASME Oil & Gas Power Conference, Washington, D. C., June 5-10, 1955. A U. S. Navy training film on the subject of shipboard vibration was presented. The film included: fundamental principles of vibrating systems; multi-mass systems; vibration excitation and response; and service problems and field investigation.

9-55 "THE DEVELOPMENT OF A HIGH-SPEED, LIGHTWEIGHT OPPOSED-PISTON ENGINE FOR SUBMARINE APPLICATION."—A. K. Antonsen. Paper No. 55-OGP-9, presented at the ASME Oil & Gas Power Conference, Washington, D. C., June 5-10, 1955. Work was started on this engine in March 1948. Two years later test on the first experimental engine was being conducted. Several submarines are now in service powered with this engine. The engine has a 6¾-in. bore and 8-in. stroke which permits a top rotative speed of 1500 rpm at 2000-ft. piston speed. Cylinder liner, crankshaft, connecting rod and piston, cylinder block, and bearings, material, design and testing are discussed. The complete working mechanism of the engine and accessories is described. Protective devices include: governor to control engine speed and an over-speed governor. The over-speed governor controls several auxiliary safety units including blower intake-air shutoff valve and a means for reducing amount of fuel delivered to fuel-injection nozzles in event back pressure becomes excessive. An air-start interlock prevents application of starting air when turning gear is engaged. Maintenance features include: handhole covers for inspection of piston and rings; removal of pistons through crankcase handhole openings; removal of lower bearings through crankcase openings; upper bearings removed after removing top cover; removal of cylinder liners after removal of upper crankshaft. All accessories were developed and tested in the form of separate units prior to test on the assembled engine. Illustrations, curves, table.

10-55 "ANALYSIS OF THE BASIC NOISE SOURCES IN THE DIESEL ENGINE."—K. R. Mercer. Paper No. 55-OGP-4, presented at the ASME Oil & Gas Power Conference, Washington, D. C., June 5-10, 1055. A typical Naval diesel engine was tested to determine causes of high noise level. This engine is described as follows: 8 cylinders in line, 2-stroke cycle, 425 bhp at 1200

rpm (full load), engine coupled to a d-c generator, firing order 1-6-5-2-8-3-4-7, air starting, attached scavenging blower. Tests indicated noise intensity increased directly with engine speed. An increase in engine speed from 600 to 1200 rpm resulted in a 10-db rise. Noise output was independent of load at any speed. The major noise source was the scavenging blower. Noise from this part was generated by directly exciting air between the two three-lobed impellers and transmitted to the surrounding air through the intake duct. A large portion of engine noise was caused by forces directly associated with reciprocating parts including the piston and connecting rod assemblies. Injectors and exhaust valves are secondary sources of noise. Combustion noise was found to be well below other noise sources. Lower engine speed with higher b.m.e.p. is likely to result in quieter operation. Gear drives excite high frequencies that are related to natural frequencies of other parts. Engine size is related to noise radiation area and structural deflections. Resonant conditions of frame and components can be influenced by prestressing of structural members. Damping of important noise makers can be of value.

11-55 "The Preparation of Residual Fuel for Motive Power."—F. H. Smith, F. P. Downing, and J. T. Costigan. Paper No. 55-OGP-1, presented at the ASME Oil & Gas Power Conference, Washington, D. C., June 5-10, 1955. Composition of residual fuels and purifying steps are discussed. Methods used in British and U. S. are presented. The U. S. system for centrifuging residuals for diesel use calls for keeping fuel in storage at temperatures high enough to facilitate pumping. Oil is then transferred through heaters. Temperature is raised to within 10° to 15°F. of the practical flash point. Oil then enters a 100 to 500 gph centrifuge where moisture and carboids are sedimented. This oil is recirculated in a day tank and further heated in the feed heater to reduce its viscosity as required by injection-nozzle design. This system has been used in municipal power plants. One report shows a 38 per cent reduction in fuel bill. Total maintenance cost over a one year period was 12 per cent of the reduced fuel bill. Thus it is apparent all operating costs can be paid off in saving in such a system and still show a profit. A marine installation shows a saving sufficient to cover total installation cost of the equipment needed to burn residual fuel in each round trip between New Orleans and Dakar. New equipment is now being designed and tested for the preparation of residual fuels. This equipment combines efficient means for sedimentation of ash with techniques for redispersion of the asphaltenes by mechanical means. References and illustrations.

12-55 "Diesel-Engine Piston and Cover Failures."—D. Fitzgeorge and J. A. Pope; "The Motor Ship," v. XXXV. n. 420, March 1955, pp 540-543. Causes of failures in pistons and cylinder covers have been investigated under direction of the Materials Subcommittee of the British Shipbuilding Association. The work was sponsored by this Association. Calculation or estimation of thermal stresses was made on combustion chamber elements. Temperature distribution was considered an impor-

tant factor. The temperature variations in pistons of two marine Diesels were studied. Included were an oil cooled piston of a single cylinder Werkspoor engine with 700°F. maximum temperature and a water cooled piston from a Doxford with a maximum temperature of 730°F. Experience shows cylinder cover cracks are usually radial or "star-type" with the unsymmetrical cover most likely to fail. One solution for preventing cracking may be to cut a radial groove in the combustion face. This remedy has been used on a Doxford piston with success. Piston failures result from: star cracks in the crown; circumferential cracks at junction of crown and skirt; circular cracks in the crown; and burnt crowns. Four failed pistons of different design, material and manufacture showed same type of failures, circumferential cracking at junction of crown and skirt and star cracking in the crown. Data presented include drawings, photographs, micrographs, material test data, curves of temperature variations in Werkspoor and Doxford pistons.

13-55 "Lost Propellers." "The Motor Ship," v. XXXV. n. 420, March 1955. p 545. A total of 29 vessels, 500 gross tons and over, lost propellers in 1954 as against 24 in 1953, according to a report of the Committee of the Liverpool Underwriters Association.

14-55 "Fuel Features Related to Operating Experiences in Motor Ships Using Low Cost Fuels."—H. F. Jones, D. Royle and R .G. Sayer. "The Institute of Marine Engineers Transactions." v. LXVII, n. 2, Feb. 1955, pp. 37-74. Total world tonnage of various types of ships in 1955 was about 93 million. Some 12 million tons were in the U.S.A. reserve fleet. Diesel powered ships totaled 29 million while steam accounted for 64 million tons. Types of engines in new ships during 1946-1953 were:

Type Engine	*No. Ships*
4 stroke	281
2 Stroke	
Single acting	1,736
Double acting	212

Four motor ship propulsion engines were studied. Ships A and B. They are single screw powered by four-cylinder Doxford opposed piston two-stroke engines with systems for burning residual fuels. Vessel C, a twin screw ship, is powered by two six-cylinder Doxford type engines. Vessel D, a tanker, is powered by one five-cylinder Doxford type engine. The operating data and results of tests with these four ships are presented in detail. They include: test bed trials; sea trials, fuel inspection, lubricating oil inspection, sludge weights and analysis; engine cleanliness and wear, ring condition and breakage, cylinder liner wear, engine performance.

Factors influencing cylinder liner wear include:

1. Type fuel used
   a. Corrosive properties

b. Abrasive properties
c. Carbonaceous properties

2. Type of cylinder lubricant used
   a. Non-detergent
   b. Detergent in various forms

3. Operating conditions
   a. Piston and liner cooling water temperatures
   b. Air intake temperatures
   c. Injector spray pattern
   d. m.e.p. and r.p.m.
   e. Relation of maneuvering to sea hours
   f. Centrifuging efficiency

4. General engine design features

5. Liner material and surface finish

Data indicate liner wear rate increases rapidly after it exceeds 0.6 per cent of cylinder diameter. Rapid rate increase probably due to:

1. Increase in number of broken rings

2. Excessive lateral movement between piston and liner surface accentuated by movement of the ship

3. Increase in quantity of blowby gas

Liner wear seemed to increase with increase in sulphur content of fuel. Greater ash content also increased liner wear. Detergent in lubricating oil had no significant bearing on wear.

15-55 "Cermets—A New Solution for Thermal Shock."—J. A. Stravrolakis. "Aero Digest," v. 70, n. 1, Jan. 1955, pp. 22-26. These materials, part metal and part non-metal are finding use as: thermocouple protection tubes, mill guides, exhaust and intake valve seats, high temperature water bearings, hot spinning tools, thread guides, nozzle diaphragm guide vanes, high temperature pressure vessels and hot upsetting anvil inserts. Cermets are grouped in three classes according to types of unions:

1. Mechanical mixtures. Examples include nickle-beryllium oxide and molybdenum-thorium oxide.

2. Interfacial metal ceramics. Processed at temperatures greater than the application range. Include chromium-aluminum oxide and silicon-magnesium oxide.

3. Solution cermets. Include zirconium-aluminum oxide and silicon-thorium oxide.

Production methods include: cold pressing, hot pressing, hydrostatic pressing, slip casting and extrusion. Data on the mechanical and physical properties of cermet compositions are not very complete. Advantages

include: better shock resistance than ceramics, good wear resistance, excellent mechanical properties at elevated temperature, low strategic material as substitutes for critical materials at high temperature, high strength-to-weight ratio. Disadvantages include: less thermal shock resistance than metals, lower ductility at all temperatures and high cost of fabrication.

16-55 "Some Operating Experiences at 950°F."—L. Baker and W. H. Falconer. "The Institute of Marine Engineers Transactions," v. LXVIII, n. 1, Jan. 1955, pp 1-13. The *Nestor* and *Nelus* are the first British built ships to operate with a steam temperature of 950°F. Boilers are of single-furnace design. Steam temperature controlled by operation of dampers to bypass gas into second stage of superheater and by balancing final steam temperature through use of an attemperator between the second and third stage of the superheater. Turbines are nozzle valve control instead of having a conventional maneuvering valve. They have been free of trouble even though gland and diaphragm clearances were small, 0.005 to 0.008 inch. Remote controlled turning gear was used to turn turbine rotors to prevent thermal distortions. Main gears have a design K of 120 on the actual contact length. Ni-Cr-Mo steel was used for secondary gear train. All teeth were hobbed and shaved. Pitting was noted on the *Nestor* gears. This finally corrected itself. Epicyclic gears were used for the turbo generator and on the feed pump as a speed increaser. Conclusions are:

1. No difficulties were experienced due to operation at 950°F.
2. Rapid fluctuations in temperature between 950° and 750°F., up and down, caused no difficulties.
3. Greatly reduced turbine clearances caused no trouble.
4. Boilers dampers caused no problems.
5. Motor driven feed pumps were trouble free.
6. Epicyclic gears, both set-up and reduction, were entirely satisfactory.
7. Controls for high water level and salinity of steam are satisfactory.

17-55 "The Future of the Marine Gas Turbine."—The Shipbuilder and Marine Engine Builder," v. 62, N. 564, May 1955. pp. 319-320. Discussion of possible means to encourage early installation of long life marine gas turbines in merchant vessels. It is vital to marine propulsion that marine gas-turbine units be operated under practical sea-going conditions. Several units would be required to conduct exhaustive research evaluation.

18-55 "Enclosures for Large Induction Motors."—M. R. Lory and J. F. Heidbreder. "Westinghouse Engineer," v. 15, n. 3, May 1955, pp. 82-85.

Proper motor enclosure increases life and assures trouble-free operation. Factors contributing to motor damage include: moisture, dirt, chemicals and oil. Types of enclosures include: drip proof, splash proof, weather protected, force ventilation, enclosed with self-ventilated, totally enclosed, fan cooled, and explosion proof. General cost comparison for typical 1000 HP induction motor with different enclosures is as follows:

Type Enclosure	*Relative Cost*	*Temp. Rise °C*
Drip-proof	100	40
Splash-proof	115-142	50
Weather-protected	125-155	50-40
Force-ventilated	108	
Self-ventilated base or pipe	125-155	50-40
Enclosed, self-ventilated with air to water cooler	159	40
Enclosed, self-ventilated with inert gas	179	
Totally enclosed, fan-cooled	205	55
Explosion-proof	226	55

19-55 "Crash Resistant Fuel Tanks Are on the Market."—William Beller. "Aero Digest," v. 70, n. 1, Jan. 1955, p. 27. Crash resistant wing fuel tank has been defined by CAA as one that is "able to contain fuel under an imposed load up to a maximum of that necessary to tear the wing from the fuselage." These tanks are bladder-shaped cells made of layers of rubberized nylon cloth. A cloth weighing 0.4 lbs./ft.^{3} might be used for a two-engine transport. This cloth would increase weight of the fuel cell by about 175 lbs. Test data are nearly complete that will permit designers to determine bladder cell strength for given impact load designs.

20-55 "Hysteresis Device: Torque Independent of Speed."—W. L. Butler. "General Electric Review," v. 58, n. 2, Mar. 1955, pp. 50-54. Constant torque devices employing hysteresis loss are finding wide application in many fields. Development of high energy permanent magnetic materials is primarily responsible for these devices. The device operates on the principle of driving a magnetic material through its hysteresis loop. The area of the hysteresis loop is a measure of energy consumed during one complete cycle of magnetism. A ring of magnetic material rotates beneath the poles of a permanent magnet in the hysteresis torque device and forms the basic arrangement of the device. Two types of applications include brakes utilizing the hysteresis effect for retarding purposes and clutches that make use of the effect as a limited-torque driving coupling. Applications include: yarn tension brakes for textile industries; manufacture of small diameter wire; production of tungsten wire for lamp filaments; manufacture of foil and treated paper; regulation of photographic film in projectors. Rapid expansion of uses for these devices is expected.

21-55 "Dropwise Condensation on a Metal Surface."—H. Hampson. "Engineering," v. 179, n. 4655, Apr. 15, 1955, pp. 464-469. Saving in weight, space and cost of condensers might be attained if dropwise, rather than film, condensation on metal surfaces could be continuously assured. A study of various factors influencing drop condensation is presented. Life of dropwise surfaces is dependent on: cleanliness of metal surfaces; promoter used; method of applying the promoter; metal and its surface finish; heat transfer rate; temperature of condensing surface; presence of non-condensable gases and the length and shape of the condensing surface. All metal surfaces need promoting agent to produce dropwise condensation. Cleanliness of a surface is indicated by filmwise condensation. Mirror polish gives longer duration of purely dropwise condensation than other finishes. Non-condensable gas increases the life of dropwise surfaces. Mixed promoters are more effective than a single substance. Life of purely dropwise condensation is reduced with increased rate of condensation. Promoters have been synthesised, and used under laboratory conditions to give the desired form of dropwise condensation for continuous periods exceeding 1000 hours.

22-55 "Oil Film Extent in Complete Journal Bearings."—J. A. Cole and C. J. Hughes. "The Engineer," v. 199, n. 5178, April 22, 1955, pp. 555-557. Direct visual observations of oil film distribution in journal bearings were made. Method of photographing oil film in 2-inch diameter bearing was used. This method utilized glass bearings and lubricating oil that was made to flouresce by ultraviolet irradiation. Numerous photographs are presented to show oil film distribution under various speed and load conditions. Tests show the full width film begins and finishes later than conventional theory indicates. Oil film is practically complete at high loads and speeds. Outlet film is indicated as consisting of filaments of oil covering only a fraction of the film.

23-55 "Gas Turbine Progress."—"The Marine Engineer and Naval Architect," v. 78, n. 939, Feb. 1955, pp. 39-40. Gas turbine manufacturers now number more than sixty. A few organizations are producing gas turbines in increasing numbers for stationary applications. Application of the gas turbine in the marine field has not been rapid. Shell Tankers have taken a major step in applying the gas turbine to marine use by their plan to replace existing machinery in the *Auris* with an improved 5,300 HP gas turbine. This is in addition to the completely new gas turbine driven tanker now being built. Mitsubishi engineers have completed and installed a 500 HP gas turbine in a ship, the Government owned training ship *Hokuto Maru*.

24-55 "Non-Metallic New Materials in Engineering."—N. J. L. Megson. "The Institute of Marine Engineers Transactions." v. LXVII, n. 3, Mar. 1955, pp. 77-92. Recent developments in the plastic field are discussed. Low pressure molding methods presented are:

1. Lay up of glass fabric or strand mat, impregnated with

a contact resin, between matching molds, which are equipped with heating devices for curing the resin.
2. Knitting sock built upon a male mold. Resin then applied to the sock and curing done with or without use of a mold.
3. Variation of "2" process to permit use of a female mold.
4. Glass fabric built up between male and female mold and resin drawn in by a vacuum.
5. Formation of tubes by winding preimpregnated glass tape or rovings on a rotating mandrel and curing by heating.

Durestos, an asbestos felt impregnated with a thermosetting phenolic resin, has been used for molding large structures including aircraft droptanks, wings and radar reflectors. This material can be molded by:

1. Sheets of material can be formed at 160°C. between matched molds under pressures of 100-150 p.s.i.
2. Sheets are laid up against a mold and a pressure applied by vacuum through a rubber bag. Material is heat cured at 160°C.
3. Hard forming on suitable mold and use of quick setting resins to build to desired thickness.

Properties of various plastics are compared with steel, aluminum alloy, titanium and magnesium-zirconium alloys. Advantages include: use of cheap molds; use of greater skin thickness; surfaces can be smooth and of any shape; materials resist corrosion; variable thickness and reinforcements are easy; temperature resistance of certain plastics better than many common alloys.

Disadvantages are: high cost; materials not suited to stress concentration; efficient joints are difficult; toughness is poor; temperature resistance of some plastics is indifferent.

25-55 "Practical Approach to Some Vibration and Machinery Problems in Ships."—T. W. Bunyan. Joint paper read before the Institute of Marine Engineers and the Institution of Naval Architects. "Transactions of the Institute of Marine Engineers," v. 67, n. 4, April 1955. pp. 99-112. Vibration problems and their treatment are discussed in two parts. Hull vibration material presented includes review of the problem in existing ships. Measurement, tuned ballasting, propeller excited vibration, design stage modifications, propeller aperature clearance, propeller forces, engine balancing, transverse rocking modes and resilient mountings are included. Under machinery vibration the following are included: reciprocating machinery, bolted connections, crankshaft stresses and alignment, top end bearing troubles, turbines and gears, claw couplings, built up gear wheels, gear noise, propellers, tailshaft stresses and alignment.

26-55 "A Comparison of Gas Turbines, Weight and Bulk as Factors in Choosing Propelling Plant for Naval Vessels."—D. F. Collins and D. W. Thomas. "Engineering," v. 179, n. 4651, Mar. 18, 1955. pp. 245-246. This is a summary of the paper "A Comparison of the Total Weight and

Bulk of In-Line Naval Gas Turbine Engines of Differing Degrees of Complexity" delivered to the North-East Coast Institution of Engineers and Shipbuilders, 11 March, 1955. Boost and cruise engines, in-line arrangements, cycle thermal efficiency, fuel consumption, division of power, fuel economy, weight , bulk, and size of deck openings are discussed. Conclusions are : (1) The plant comprising two double compound intercooled engines, with the cruise engine using a heat exchanger, gives the lowest plant-fuel weight for a medium size naval vessel. Deck openings are smallest for this plant. (2) The plant with two single compressor engines, with heat exchanger on the cruising engine, is shortest in length, second best in total weight, but is among the worst for size of deck opening. It is most difficult to install. (3) The plant with two intercooled double-compound engines has lowest value of total bulk, about equals the best for size of deck opening but is among the worst from point of total weight. (4) The plant with double-compound engine, without inter-cooling or heat exchange, is poorest in respect to plant-fuel weight, bulk and size of deck openings. Generally the most attractive plant is the one with two double-compound intercooled engines with heat exchange on the cruising engine.

27-55 "The Development of a 500 HP Multipurpose Gas Turbine Engine."—P. G. Carlson and I. M. Swatman. Presented at the ASME Diamond Jubilee Spring Meeting, Baltimore, Md., April 18-21, 1955. Paper number 55-S-46. A 500 HP gas turbine with multi-purpose features has been developed. The new unit has a design life of 5,000 hours. It is based on an earlier turbine, T-400 J, which was installed as an emergency generator drive in the *Timmerman* in January 1951. The new unit can be made constant or variable speed. The constant speed version is the Jupiter T-520 J and the variable speed the Jupiter T-522 J. Design, development and test of the new units were conducted in a relatively short time. In the first year of test 1000 hours operation were achieved. Some 720 hours were logged on one unit. In the second six months of test 900 hours of test time was recorded. Based on test and operation to date it appears that the Jupiter power plant can achieve 5000 hour design life at a continuous rating of 420 HP.

AN OUTLINE OF ATOMIC PHYSICS

By Oswald H. Blackwood, Thomas H. Osgood, Arthur E. Ruark

Third Edition

Published by John Wiley & Sons, Inc., 1955

501 Pages

Reviewed by Associate Professor E. J. Cook

Department of Electrical Engineering, U. S. Naval Academy

This is an excellent book for the purpose for which it is intended, namely to give a non-specialist having a background of general physics a grasp of the fundamentals of atomic physics. The authors have made a judicious choice of the more useful and important topics in this edition and in so doing they have not lost the unity or teachability of the earlier editions. Moreover, the new material does not appear to be "tacked on" as is so often the case where authors augment a text book to keep pace with the recent advances in the field.

The emphasis throughout this book is on the basic principles and interpretations rather than on minute details, however, ample references are given to the original literature or more complete works for the more serious students in this field.

An added feature of this book is the inclusion of starred paragraphs which may be omitted without seriously detracting or destroying the continuity of the presentation. Roughly, two-thirds of this text is devoted to atomic physics—structure of atoms, molecules and the nature of radiations. The latter portion treats the fundamental particles, the nucleus of the atom, and includes an up-to-date discussion of nuclear magnetic resonance, nature of cosmic rays, and nuclear physics applications.

The large number of well-chosen problems with answers at the end of each chapter and the clear-cut illustrations and tables contribute to the value of this book.

ELEMENTARY THEORY OF NUCLEAR SHELL STRUCTURE

By M. G. MAYER and J. H. D. JENSEN

Published by John Wiley Sons, Inc., 1955

$7.75

Reviewed by Assistant Professor E. E. Gardner

Department of Electrical Engineering, U.S. Naval Academy

The authors have presented in this monograph a detailed qualitative discussion of the most successful model yet devised for the nuclei of the atoms, namely, the nuclear shell model. This model is quite similar in many respects to the model that has been used to describe atomic structure. There is one principal difference, however, in that in considering atomic structure one deals only with particles bearing an electric charge—a positively charged nucleus and negatively charged electrons—whereas in the nucleus one considers the interactions of charged particles, protons, and uncharged particles, neutrons.

The nuclear shell model is based on assumptions essentially the same as those inherent in the Bohr-Pauli explanation of the periodic table: (1) The possible "nucleon orbits" are characterized by a set of quantum numbers (n, l, j, m) and the corresponding levels are successively filled with protons and neutrons. (2) Each proton level of quantum number j can contain not more than $2j+1$ protons; the same holds for neutrons. This model of the nucleus has been exceptionally successful in accounting for a large number of properties of a vast majority of the nuclei. The phenomena related to the "magic numbers" are explained in terms of closed shells. In all cases where the nuclei contain nearly closed shells (closed shell plus or minus one nucleon) the experimental and predicted values of the angular momenta agree exactly. Furthermore, the nuclear model has been successful at predicting the spin and parity of nuclei and the sign of the quadrupole moment. Since little is known about the nuclear potential, agreement between the calculated and experimental values of the electric and magnetic moments has not been too satisfactory.

The authors state in the preface "as readers we had in mind physicists and chemists whose daily work concerns the nucleus"—this statement should have been underlined since this monograph is not in any sense a textbook for the general student of physics or chemistry. The reviewer found the reading of this work tedious and somewhat disappointing. He was left with the impression that much needless repetition prolonged the length of this monograph and too brief a treatment of the necessary mathematics impaired its usefulness as an introduction to nuclear physics.

BOOK REVIEW

ELECTRONS, ATOMS, METALS AND ALLOYS

By William Hume-Rothery
Published in 1948
Revised and reprinted 1955
Published by Philosophical Library, Inc.
15 East 40th St., New York 16, N.Y.
and Iliffe and Sons, Ltd.
Dorset House, Stamford Street, London, S.E. 1
387 pages and illustrations
Reviewed by CDR C. N. Payne, USN
Department of Marine Engineering, U.S. Naval Academy

In a novel format for a modern technical work, this book is presented as a dialogue between technical men, characterized by "Older Metallurgist" and "Young Scientist." The purpose is a survey of the modern theory of the structure and properties of metals and alloys. It deals entirely with the science of metallurgy as developed from basic atomic theory, rather than the empirical science or the art of metallurgy, to which so many engineering textbooks are limited.

The approach of the book is epitomized in the following bit of dialogue:

Young Scientist: ". . . at the present time we are only at the beginning."

Older Metallurgist: "That's nonsense. Metallurgy is one of the oldest of all the sciences."

Young Scientist: "On the contrary, it is one of the oldest of arts, but one of the youngest of sciences."

To give the necessary background to the technical, but non-scientific reader, a review of the basic theory of the atomic structure of matter is presented in the first part of the book. Intricate mathematics is avoided in a very informative discussion of the wave and particle nature of matter as treated by the theories of wave and quantum mechanics.

In Part II, the dialogue progresses to a discussion of the nature of a metal, i.e., the behavior and properties of an aggregate of atoms, crystalline structure, free electron theory, valence, magnetism, etc. Surprisingly, "Older Metallurgist", unschooled in the modern theory, feels more and more at home with "Young Scientist" and keeps up his end of the dialogue in an admirable fashion.

The dialogue turns, in Part III, to a discussion of the nature of an alloy as seen in light of the background of the first two parts of the book. The rational basis of various equilibrium diagrams is explained; electron compounds and superlattice structures are discussed.

In Part IV the discussion centers about the structure of the atomic nucleus, radioactivity, fundamental particles, and nuclear fission.

In these four parts the survey is completed. The author makes no claims that any phase of the subject is completely covered. He has listed references at the end of each chapter, where pertinent, for more complete information on the various topics.

Every engineer who is concerned with materials will find this book extremely interesting and stimulating. It is useful also to the reader who wishes a brief description of modern atomic theory, preparatory to further studies in the field of nuclear engineering.

BOOK REVIEW

SERVOMECHANISMS AND REGULATING SYSTEM DESIGN, VOL. II

By HAROLD CHESTNUT and R. W. MAYER

Published by John Wiley & Sons, Inc., 1955

368 pages

$8.50

Reviewed by Associate Professor J. A. Lee, Jr.

Department of Electrical Engineering, U.S. Naval Academy

In recent years there have been many books written on the subject of automatic control or servo-systems, but for the most part these have been confined to the fundamental principles of analysis of linear systems. Design considerations involving the interpretation of system requirements into suitable hardware terms are generally omitted or are treated only sparingly. With the appearance of Volume II of Chestnut and Mayer's servomechanisms series, a long step toward correcting this situation has been taken.

The sequence in which the material is presented is such that the book is suitable for the advanced graduate student as well as the practicing engineer. In addition to serving as a review of fundamental principles, Chapter I makes clear at the outset the relation between these and the problems encountered in the design of servo systems and automatic regulators. This is done by discussing first the subject of measurements and measurement techniques. Not only is this sound pedagogically, since it permits laboratory work to begin immediately, but practically speaking all design based on analytical work must be thoroughly checked by measurements to determine if it meets the requirements of the job.

Of great importance to the servomechanist is the problem of system error as caused by reference input variations and disturbances or extraneous signals. This is treated at length. Methods are indicated for determining quickly the error resulting from specific inputs and examples are given illustrating these. Methods are developed which afford a quantitative description of the effects of "noise" in a system. Finally, an example is given illustrating the process of optimizing system design to obtain a satisfactory balance of errors, systematic and noise.

In the chapters which follow many more important topics are discussed, such as,

(1) Selection of power elements.
(2) Stabilizing networks, with charts showing various networks, their transfer functions and gain vs. frequency characteristics, criterion for choice and its use in the loop.
(3) Amplifier design for servo-systems.
(4) All a-c servo operation.
(5) Theory applied to non-linear servomechanisms.

As one can observe from the foregoing, the scope of the book is considerable but at the same time, in the opinion of this reviewer, the treatment is adequate. Not to be overlooked is the rather comprehensive bibliography chronologically arranged.

STEAM, ITS GENERATION AND USE

Thirty-seventh edition, 1955

Babcock and Wilcox Company

Reviewed by Assistant Professor J. H. Smith,

Department of Marine Engineering, U.S. Naval Academy

This book thoroughly covers the field of steam generation. It is well written, and will be of value to both operating and design engineers. It could well be used as a supplementary or reference text for college engineering courses at the senior level.

The coverage, from a broad viewpoint, includes:

(1) Historical items of interest
(2) Reserves, production, and consumption of fuels in the world and the United States
(3) Fundamentals of thermodynamics, fluid flow, heat transfer, and metallurgy as applied to steam generation and generating equipment
(4) Design, types, uses and construction of boilers, furnaces, economizers, air heaters, superheaters, and reheaters; and their care and operation
(5) Operating procedures for different types of coal handling equipment
(6) Paper making, development of alkaline pulping, chemical and heat recovery steam generators
(7) Economics in selection of steam producing equipment
(8) Nuclear reactors
(9) Marine steam generating equipment

The basic theory and principles of thermodynamics, heat transfer, and fluid flow are presented in a manner easily understood by the undergraduate engineering student. These fundamentals are treated in a complete but concise fashion.

Design equations, including experimental data, for boilers and boiler components are given. Numerous example calculations are shown which make the book of particular utility to the student.

GEOLOGY IN ENGINEERING

JOHN R. SCHULTZ and ARTHUR B. CLEAVES

Published in 1955

John Wiley & Sons, Inc.

592 pages. $8.75

This book was designed as both a text and a practical reference. It integrates geology and engineering principally for the benefit of the engineer. It will not make a geologist out of an engineer because of the necessity for compressing so much in so little space. It should improve communications between engineers and geologists and should induce the former to make optimum use of the services of the latter before becoming committed to a "house built on sand."

SHIP AND BOAT BUILDER ANNUAL REVIEW 1955

compiled by

"Ships and Boat Builder"

Published in London in 1955

John Trundell, Limited

380 pages. Price in London: 30 Shillings

With a foreword by Major L. Cocollis, Retiring President

Ship and Boat Builders National Federation

This book contains 13 articles on such subjects as "Trends in Tug Propulsion," "Small Craft of the NATO Navies," "New Marine Equipment in 1954," and "The Small Steam Engine: Its Place in the Modern Marine World."

The main body of the book is devoted to a directory of Boat Builders in Great Britain and Overseas, a section on specifications of engines, one on "Materials, Fittings and Equipment" and additional directories of Consultants (Great Britain); Associations (Trade, Learned Societies and Miscellaneous); British owners of small craft; Dock and Harbor Authorities of Great Britain and Ireland; Commercial Diplomatic Officers and Trade Commissioners; and British Chambers of Commerce.

MARTIN L. KATZENSTEIN

On July 21, 1955, in New York City, Martin L. Katzenstein died at the age of 75.

Mr. Katzenstein was one of the early Associate Members of the Society, having been a member since January 1, 1902. A graduate of City College of New York and Cornell University, he was associated for twenty-five years with The Worthington Corporation, where he headed the marine department. In 1928, he organized the Warren Engineering Company, and was also the Eastern marine representative of The Warren Steam Pump Company.

CHANGES IN MEMBERSHIP

The Society is very much pleased to report that the following have become members since the publication of the May, 1955, issue of the JOURNAL:

NAVAL

Fletcher, Robert C., Commander, USN,
USS DONNER (LSD 20) c/o Fleet P.O., San Francisco, Calif.
Iverson, Willard Marcus, Commander, USN,
Inspector of Naval Material, Treasure Island, San Francisco, Calif.
Kelty, Kenneth Gile, Lieutenant, USNR,
Engineer, Atomic Power Equipment Dept.
General Electric Company
Mail: 2900 Campbell Ave., Schenectady, N. Y.
Kraft, Wendell E., Captain, USN, Ret.,
Asso. Professor of Engineering,
Trinity College, Hartford, Conn.
Mail: 29 Smallwood Road, West Hartford, Conn.
Porter, William Robert, Lieutenant, USN,
412 South Parton St., Santa Ana, Calif.
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ERRORS IN LISTING CHANGES IN MAY, 1955, JOURNAL

It is regretted that the following errors were made in listing Changes in Membership in the May, 1955, issue of the JOURNAL.

Lieut. Comdr. John Kent Husion, USNR, should have been listed as Chief Engineer, Marietta Manufacturing Co., instead of Henrietta Manufacturing Company.

Civil member listed as "James B. Stack", should have been listed as John Berchmans Stack, and his address as 131 Poplar Street, instead of 31 Poplar Street.

The Society has been informed, to its very great regret, of the death of the following members since the publication of the May, 1955, Journal:

Brunner, Harry Charles, Civil Member

Connelly, L. A., Civil Member

Katzenstein, Martin L., Associate Member

La Conga, Jack F., Life Member (Naval)

Murphy, Moyes J., Naval Member

ASSOCIATION NOTES

ORIGINAL ARTICLES

The editor of the JOURNAL is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $50.00 to $250.00 depending upon length, interest and professional value.

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. Include on a separate page a short biographical sketch(es) of the author(s). 50 to 100 words for each author is desired.

Manuscripts should be addressed to:

Secretary-Treasurer
American Society of Naval Engineers
605 F St., N. W.
Washington 4, D. C.

Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 10 reprint copies of his article free of charge. Additional copies may be purchased from the Society if the request is received 30 days prior to the publication date which is the 25th of the issue month. Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Manuscripts which are accepted will be published in the earliest JOURNAL which has not yet been closed (60 days before publication) and for which sufficient material is not already on hand.

PHOTOGRAPHS

Photographs of current or historical interest to readers of the JOURNAL are desired. Any which are accepted by the Society will be purchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

NOTICE OF DEATHS

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be appreciated greatly if any member who learns of the death of a member will advise the Secretary-Treasurer.

SOCIETY LAPEL BUTTONS

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold background.

It is available to all members at fifty cents (50c) each.

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a) The article is reprinted in full. Extracts or summaries or condensations may be printed with the consent of the author.

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d) If the author is a military officer or a civilian employe of the Department of Defense, the following note shall be carried:

> "The views expressed herein are the personal opinions of the author and are not necessarily the official views of the Department of Defense or of a Military Department."

FINANCIAL STATEMENTS, 1954

STATEMENT OF INCOME AND EXPENDITURE

INCOME

Publication:		
Dues	$22,842.49	
Advertisements	6,781.25	
Sales	730.34	
Subscriptions	14,541.50	$44,895.58
Interest on Investments		1,500.00
Banquet, 1954		2,178.39
		$48,573.97

EXPENDITURES

Publication:		
Printing	$24,391.77	
Engraving	2,269.01	
Manuscript	4,395.00	
Postage	681.19	
Commission	400.75	
Exchange	.81	$32,138.53
Salaries		10,200.00
General Expense		2,816.58
Bad debts & Miscellaneous		1,611.36
		$46,766.47
NET GAIN		$ 1,807.50

BALANCE STATEMENT

Assets December 31, 1954:

Cash		$ 6,114.09	
Petty Cash		50.00	
Accounts Receivable:			
Advertisments	$ 3,238.20		
Dues	1,570.22		
Sales	567.48		
Subscriptions	682.50	6,058.40	
Investments:			
U.S. Defense Bonds, Series G	$58,000.00		
Washington Gas Light Co. Bonds	997.50	58,997.50	
Furniture		237.00	
Manuscripts paid in advance		1,260.00	
Banquet, 1955, paid in advance		100.00	$72,816.99
Liabilities (Accounting Only):			
Dues paid in advance	$ 313.36		
Subscriptions paid in advance	7,437.25		
Advertisements paid in advance	362.50		
Brand Award	59.00		8,172.11
Net Assets December 31, 1954			$64,644.88
Total Assets January 1, 1954			62,837.38
NET GAIN			$ 1,807.50

Respectfully submitted,
/S/ ROBERT B. MADDEN
Acting Secretary-Treasurer

Audited and Found Correct

/S/ CHARLES G. COOPER
/S/ M. D. GIBSON, JR.
/S/ J. B. ARMSTRONG
Audit Committee

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1896–97 P. A. Engineer F. C. BIEG, U. S. Navy
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1918 Lt. Comdr. F. W. STERLING, U. S. Navy, Retired
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1924–25 Commander BRYSON BRUCE, U. S. Navy
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1927 Commander H. B. HIRD, U. S. Navy
1928 { Commander H. B. HIRD, U. S. Navy / Captain O. L. COX, U. S. Navy }
1929–30 Commander H. T. SMITH, U. S. Navy
1931 Captain O. L. COX, U. S. Navy
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1933–34 Commander H. B. HIRD, U. S. Navy
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1936 { Commander C. S. GILLETTE, U. S. Navy / Commander ROGER W. PAINE, U. S. Navy }
1937 Commander ROGER W. PAINE, U. S. Navy
1938 { Commander ROGER W. PAINE, U. S. Navy / Lt. Comdr. GUY CHADWICK, U. S. Navy }
1939–40 Lt. Comdr. GUY CHADWICK, U. S. Navy
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